AD-A242 493

High-Temperature Combustor And Seal For a Water Piston Propulsor

Final Report

91-15351

CATERPILLAR

Solar Turbines

Solar Turbines Incorporated

91

UNCLASSIFIED
SECURITY CLASSIFICATION OF THIS PAGE

REPORT DOCUMENTATION PAGE

1a. REPORT SECURITY CLASSIFICATION: UNCLASSIFIED	1b. RESTRICTIVE MARKINGS
2a. SECURITY CLASSIFICATION AUTHORITY	3. DISTRIBUTION/AVAILABILITY OF REPORT: Approved for public release, distribution is Unlimited
2b. DECLASSIFICATION/DOWNGRADING SCHEDULE	
4. PERFORMING ORGANIZATION REPORT NUMBER(S): SR90-R-5253-76	5. MONITORING ORGANIZATION REPORT NUMBER(S): DTRC-SD-CR-16/91
6a. NAME OF PERFORMING ORGANIZATION: SOLAR TURBINES INCORPORATED; 6b. OFFICE SYMBOL (if applicable)	7a. NAME OF MONITORING ORGANIZATION: DAVID TAYLOR RESEARCH CENTER
6c. ADDRESS (City, State, and ZIP Code): P.O. BOX 85376, 2200 Pacific Highway, San Diego, CA 92186-5376	7b. ADDRESS (City, State, and ZIP Code): Code 1240 (MCPO), Bethesda, MD 20084-5000
8a. NAME OF FUNDING/SPONSORING ORGANIZATION: MCRDAC - AWT; 8b. OFFICE SYMBOL (if applicable)	9. PROCUREMENT INSTRUMENT IDENTIFICATION NUMBER: N00167-85-C-0042
8c. ADDRESS (City, State, and ZIP Code): Quantico, VA 22136-5080	10. SOURCE OF FUNDING NUMBERS

PROGRAM ELEMENT NO	PROJECT NO	TASK NO	WORK UNIT ACCESSION NO
62131M	C3150		DN978568

11. TITLE (Include Security Classification)

HIGH TEMPERATURE COMBUSTOR AND SEAL FOR A WATER PISTON PROPULSOR

12. PERSONAL AUTHOR(S)
M.A. Galica and R.T. LeCren

13a. TYPE OF REPORT	13b. TIME COVERED	14. DATE OF REPORT (Year, Month, Day)	15. PAGE COUNT
Final	FROM 06/85 TO 03/91	91/03	

16. SUPPLEMENTARY NOTATION

17. COSATI CODES			18. SUBJECT TERMS (Continue on reverse if necessary and identify by block number)
FIELD	GROUP	SUB-GROUP	Combustor
			Water piston propulsor

19. ABSTRACT (Continue on reverse if necessary and identify by block number)

The detailed design and testing of a high-temperature combustor and seal for a Water Piston Propulsor is reported. The system is comprised of a high-pressure combustor together with a carbon-graphite rotor-contacting face seal. Analyses are presented to support the design of an air-cooled combustor liner. The seal is pneumatically loaded against the rotor face using a variable pressure. Superalloy materials are selected for most components with a vitreous enamel corrosion-resistant coating applied to the high-temperature combustor liner.

The combustor was subjected to a series of rig tests to define its performanceprior to integrating it with the rotor system. A simulated rotor test was also conducted to study the seal plate bellows pressure relationship required to attain a satisfactory seal. After the Solar combustion and seal system was integrated with the rotor system designed and fabricated by Tracor Hydronautics, three series of water channel tests were conducted to determine the overall system performance. The combustor performed satisfactorily although the imposed operating conditions, i.e. pressure and air flow, were significantly different from the original design specifications, requiring the combustor to operate at higher combustor loadings.

20. DISTRIBUTION/AVAILABILITY OF ABSTRACT: ☐ UNCLASSIFIED/UNLIMITED ☒ SAME AS RPT ☐ DTIC USERS	21. ABSTRACT SECURITY CLASSIFICATION: UNCLASSIFIED
22a. NAME OF RESPONSIBLE INDIVIDUAL: RICHARD SWANEK	22b. TELEPHONE (Include Area Code): (301)227-1852; 22c. OFFICE SYMBOL: MCPO

DD FORM 1473, 84 MAR — 83 APR edition may be used until exhausted. All other editions are obsolete.

SECURITY CLASSIFICATION OF THIS PAGE

The tested propulsion efficiency of the WPP was lower than predicted. An attempt to improved performance by matching the shape of the combustor outlet to the roto passage inlet yielded minimal improvements.

High-Temperature Combustor And Seal For a Water Piston Propulsor

Final Report

Prepared By:

M.A. Galica and R.T. LeCren
Solar Turbines Incorporated
P.O. Box 85376
San Diego, Ca. 92186-5376

For:

David Taylor Research Center
Bethesda, Maryland 20084-5000
Attn: R. Swanek, Code 1124
Under Contract No. N00167-85-C-0042

A-1

SR90-R-5253-76
March 1991

TABLE OF CONTENTS

LIST OF TABLES

ABSTRACT

The detailed design and testing of a high-temperature combustor and seal for a Water Piston Propulsor is reported. The system is comprised of a high-pressure combustor together with a carbon-graphite rotor-contacting face seal. Analyses are presented to support the design of an air-cooled combustor liner. The seal is pneumatically loaded against the rotor face using a variable pressure. Superalloy materials are selected for most components with a vitreous enamel corrosion-resistant coating applied to the high-temperature combustor liner.

The combustor was subjected to a series of rig tests to define its performance prior to integrating it with the rotor system. A simulated rotor test was also conducted to study the seal plate bellows pressure relationship required to attain a satisfactory seal.

After the Solar combustor and seal system was integrated with the rotor system designed and fabricated by Tracor Hydronautics, three series of water channel tests were conducted to determine the overall system performance. The combustor performed satisfactorily although the imposed operating conditions, ie. pressure and air flow, were significantly different from the original design specifications, requiring the combustor to operate at higher combustor loadings.

The tested propulsion efficiency of the WPP was lower than predicted. An attempt to improve performance by matching the shape of the combustor outlet to the rotor passage inlet yielded minimal improvements.

1.0

INTRODUCTION

The desire to develop U.S. Marine Corps amphibious vehicles with higher in-water speeds has led to the advancement of the highly innovative Water Piston Propulsor (WPP) concept. The significant advantages of this concept include lower weight, which translates into a lower horsepower requirement per ton of vehicle, and the elimination of additional armor as the WPP systems are largely external to the hull.

Tests had demonstrated the validity of existing analytical thrust prediction codes through single channel experiments at low and intermediate temperatures. The next phase of a program necessary to advance the WPP concept to operational readiness was to evaluate high-temperature operation of multi-passage systems in water channels and towing tanks. These tests were to be followed by a full vehicle test. To this end, a suitable high-temperature combustor and seal system was required to be integrated with the WPP rotor. The definition, fabrication, and acceptance testing of such a high-temperature combustor and seal system was the goal of Solar's program.

There remained, however, significant development risks associated with the advancement of the WPP concept. These included the stability of, and heat transfer across, the gas/water interface; the achievement of satisfactory combustion stability when operating in concert with potentially fluctuating downstream pressure conditions; and the definition of a seal system with acceptable leakage and parametric power characteristics over the life cycle of the unit.

The objectives therefore, were to provide a high-temperature combustor and seal system that addressed the critical problem areas of thermal shock resistance, hot corrosion, and combustor performance and to support the water channel tests of the complete system.

2.0

SUMMARY

As a participant in the program to evaluate the performance of the Water Piston Propulsor concept, Solar, designed, fabricated, and tested a combustor and seal assembly including a control system. The combustor and fuel injector design were based on small gas turbine background. The seal design was based on rubbing seal experience. the combustor was subjected to a series of rig tests to confirm that the performance criteria were met. These criteria included operation over a range of airflows, pressures, and inlet temperatures with an overall pressure of 7% and an outlet temperature of 3000°F. It was demonstrated that the open loop fuel control system could control the outlet temperature to 3000°F $\pm$ 200°F. Pressure drop and efficiency goals were met.

In order to investigate the seal loading parameters before the water channel tests started, a series of tests was run using a simulated rotor. The purpose was to select a control approach. The results were used in the tests in the water channel.

The Solar combustor and seal system were then integrated with the Tracor rotor assembly and installed in a recirculating water channel facility at Tracor. While the overall propulsion efficiency of the WPP was disappointingly low, the combustor functioned effectively even though the operating conditions imposed by the rotor system were different than designed for. The actual pressure/flow relation caused the combustor to operate at higher combustion loadings and velocities than those designed for.

Solar attempted to analyze the reasons for the poor propulsion efficiency using a computational fluid dynamics code (Phoenics). The results of this work indicated that the gas was preferentially escaping down the leading edge of the channel, i.e. the gas to water interface was not stable, and that this effect could be avoided with an increased number of channels in the rotor.

3.0

DESIGN

3.1 DESIGN SPECIFICATIONS

Solar was fully responsive to the initial required technical specifications of the combustor/face seal/fuel controller assembly, with the exception of item 3.4 of the contract, Combustor Physical Constraints. The initial specification stated that the combustor have a diameter and length of no more than 4.0 and 12.0 inches, respectively. Solar's design is larger than the stated dimensions with a maximum diameter and length of 6.0 and 11.7 inches, respectively. It should be noted, though, that the 6.0 diameter is due to a flange placed at the rear of the combustor. This flange will facilitate ease of disassembly for inspection purposes. When the combustor assembly enters production phases, the maximum diameter could be reduced to 4.51 inches.

The following technical specifications are those required of the WPP. The specifications are identical to that of those listed in Solar's contract.

3.1.1 Combustor Inlet Air Conditions

The combustor is capable of accepting an inlet air flow of 1.1 pounds per second (pps). The temperature of the inlet air flow can vary from 200°F (i.e., when a compressed gas storage tank is used as a gas source in conjunction with a gas heater for initial testing) to 400°F when a compressor is used as the gas source. The inlet pressure can range from 50 psi (i.e., when the engine driving the compressor is idle) to 225 psi at full engine speed. The combustor is capable of remaining lit over these inlet pressure and temperature extremes. In summary:

Combustor Air Supply Pressure: 50-225 psi (externally regulated)
Combustor Air Supply Temperature: 200-400°F

3.1.2 Combustor Fuel Supply Conditions

The fuel supply to the combustor is sufficient to raise the mean gas temperature at the combustor discharge to 3000°F. It was assumed that the inlet air is comprised of the standard percentage of atmospheric gases. The inlet air pressure was assumed to vary from 50 to 225 psia as a function of diesel engine speed. The fuel supply pressure is sufficient to provide the required amount of fuel for near-stoichiometric combustion over the given range of inlet air pressures and temperatures. The fuel used was diesel (MIL-F-16884) with a specific heat capacity of approximately 19,000 Btu's/lb.

3.1.3 Fuel Controller Operation

The fuel controller is capable of regulating fuel supply pressure in response to variations in air supply pressure and temperature. Solar determined the correct relationship between fuel pressure, air pressure and air temperature for the given range of inlet air conditions. It was anticipated that the diesel engine throttle would serve as the thrust regulator with the fuel controller simply maintaining a proper fuel/air mixture as a function of inlet air pressure and temperature. Solar included, as part of the combustor design, sensors that determined inlet air temperature and pressure.

3.1.4 Combustor Physical Constraints

The combustor is roughly cylindrical in shape. The maximum external diameter of the combustor is 6.0 inches, with a maximum overall length of 11.7 inches. These dimensions include all required cooling passages, fuel nozzles, air supply passages, and associated sensors.

3.1.5 System Operational Considerations

The system requirements varied as a function of vehicle operational situations. The following procedures are representative on an in-water shutdown, in-water start-up, in-air start-up and a water to land egress/shutdown.

3.1.5.1 In-Water Shutdown

a. Operator pulls back on the engine throttle thereby reducing the air supply pressure.

b. The contractor supplied algorithm (electrical, digital or fluidic) regulates fuel supply pressure and hence fuel flow to the fuel nozzle in response to changes in the air supply pressure and temperatures.

c. Vehicle proceeds at low speeds (i.e., low thrust associated with approximately 50 psi supply pressure).

d. For unspecified reasons, the vehicle is required to shutdown its in-water propulsor while water-borne.

- Operator sends an electrical signal to the fuel controller indicating that a shutdown is desired.

- Fuel controller suspends fuel flow to the nozzle.

- Nozzle is purged with air to prevent fuel from distilling and clogging the nozzle. (Note: the air compressor is continuing to supply 50 psia, thereby cooling the combustor).

- The air compressor is de-clutched from the engine and sea water floods the combustor, its air supply passages and its enclosure sensors.

3.1.5.2 In-Water Start-Up

The amphibian vehicle may be required to restart its propulsors while waterborne. This requirement occurs after the vehicle has powered itself down the ramp of an amphibious ship, has powered itself down to and into the ocean or is required to restart its propulsor following an in-water shutdown. The following steps were anticipated:

- The diesel engine is running at idle and the clutch to the air compressor is engaged.

- Air flows to the combustor and purges salt water from the combustor and its surrounding passages.

- The diesel engine speed will be increased to provide a Solar specified inlet air pressure for ignition.

- The operator shall initiate lite-off by sending an electrical signal to the fuel controller and igniter. The fuel controller shall initiate fuel flow and fire the igniter.

- The operator shall bring the diesel engine throttle up to full power, thereby causing the vehicle to accelerate.

3.1.5.3 In-Air Start-Up

This condition is not allowed. Although the compressor may be operated without fire in the combustor, the rotor walls would damaged if the combustor is fired without the cooling influence of the surrounding water.

3.1.5.4 Land Egress/Shutdown

As the vehicle passes through the surf zone, the tracks would deployed and turning. When contact with the beach is made, the tracks would pull the vehicle forward; thereby, preventing pitchpoling and/or the washing of the vehicle back out into the surf. During this period, the engine would be operating at some mid-power level. As the vehicle performs this maneuver, the combustion components would undergo the following sequence:

- In the surf zone, the combustor shall remain lit.

- When contact with the beach is firmly made, the operator shall initiate propulsor shutdown by sending an electrical signal to the fuel controller in 3.1.5.1, the fuel controller shall shutdown fuel flow and purge the nozzle with air.

- Following a combustor cool-down period, the operator shall declutch the compressor from the diesel engine and stow the transom flap as he progresses up the beach.

3.1.6 Igniter Operations

As noted in Section 3.1.5, the igniter would used only when the propulsor is fully submerged after air flow through the combustor has dried the internal components. Solar has specified the inlet air pressure and fuel supply rate at which the igniter is designed to operate.

3.1.7 Corrosion Resistance

As described in the operational sequences given above, the combustor and its associated components and sensors would not normally be submerged while hot. However, a diesel engine or compressor failure would allow the combustors and their associated components to be flooded while hot. Under these conditions, it is desired that all components survive the thermal shock and corrosive effects. Furthermore, the combustor system will be fully flooded whenever the propulsor is shutdown in-water. Normal operating procedures require compressor start-up prior to submergence and compressor shutdown after emergence from the water thereby preventing complete submergence. The atmosphere in all cases is salt laden and highly corrosive effects have been anticipated.

3.1.8 Seal Design Requirements

Several seals have been conceptually designed for providing the interface between the stationary combustor and the water-filled rotor. Preliminary concepts are illustrated in Figure 1 (A-C). Each concept was somewhat dependent upon the availability of materials with specific tribophysical and thermal characteristics. The efficiency of the WPP concept is highly dependent upon the effectiveness of the seal, both in preventing gas leakage and in minimizing the rotational torque loss. If the rotor is to provide a steady speed for a given exit angle of each channel, the drag associated with the seal must be relatively constant over hundreds of hours of use. As shown in Figure 1, the seal strategy cannot be separated from the rotor design. However, the rotor design is somewhat more flexible since its surfaces are periodically cooled. Thus, a decision was made to seek appropriate seal technology before finalizing rotor design specifications. Solar specified associated rotor design requirements but was not responsible for the final rotor design. In summary, the seal was required to meet the following criteria.

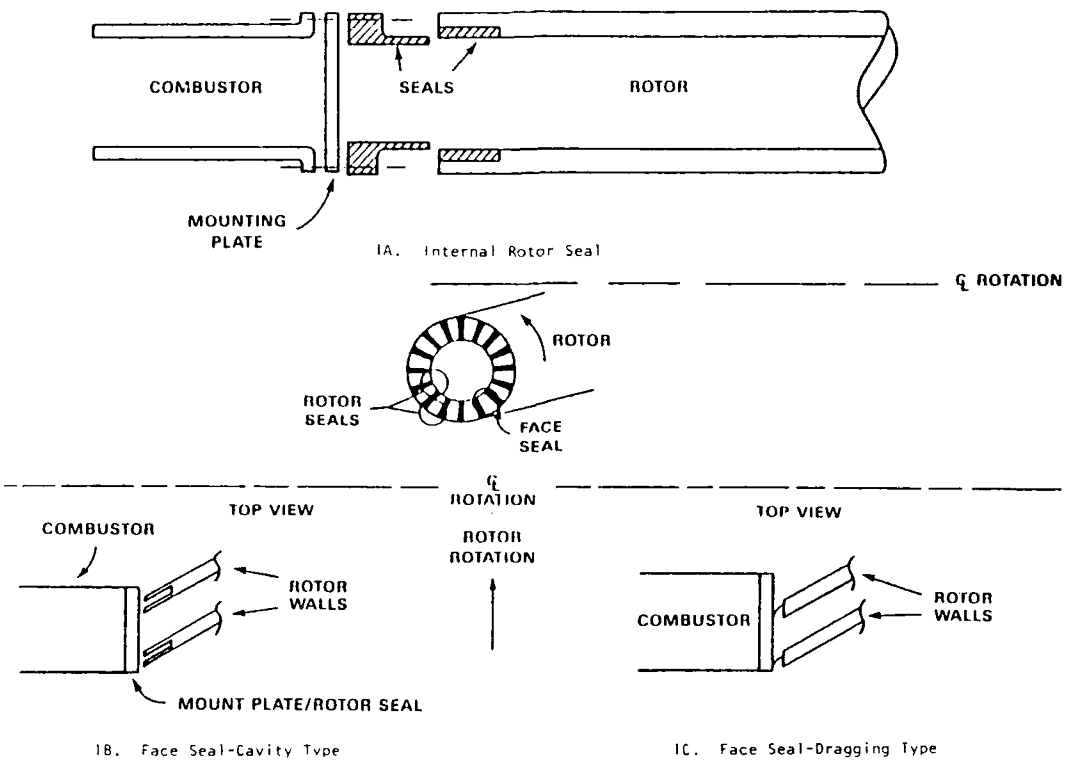


Figure 1. Preliminary Concept of Rotor with Face Seal Options

3.1.8.1 Seal Thermal Shock

The seal system is in contact with 225 psi air at 3000°F at one segment, while concurrently being exposed to 40°F sea water at another segment. Similarly, a single piece of seal would alternately be exposed to 300°F gas and 40°F water. The seal system was required to resist cracking, fracture, chipping, accelerated corrosion, and accelerated wear under such rapid thermal cycling in a corrosive medium.

3.1.8.2 Rotor Geometry

The upstream end of the rotor is not fixed and can be conceptually altered within limits to accommodate a seal design. The following specifications pertain to the rotor shown in Figure 2.

Length: 32 inches
Diameter: 16 inches OD
Number of Channels: 16
Entrance Helix Angle: 40.9 degrees
Exit Helix Angle: 14.2 degrees

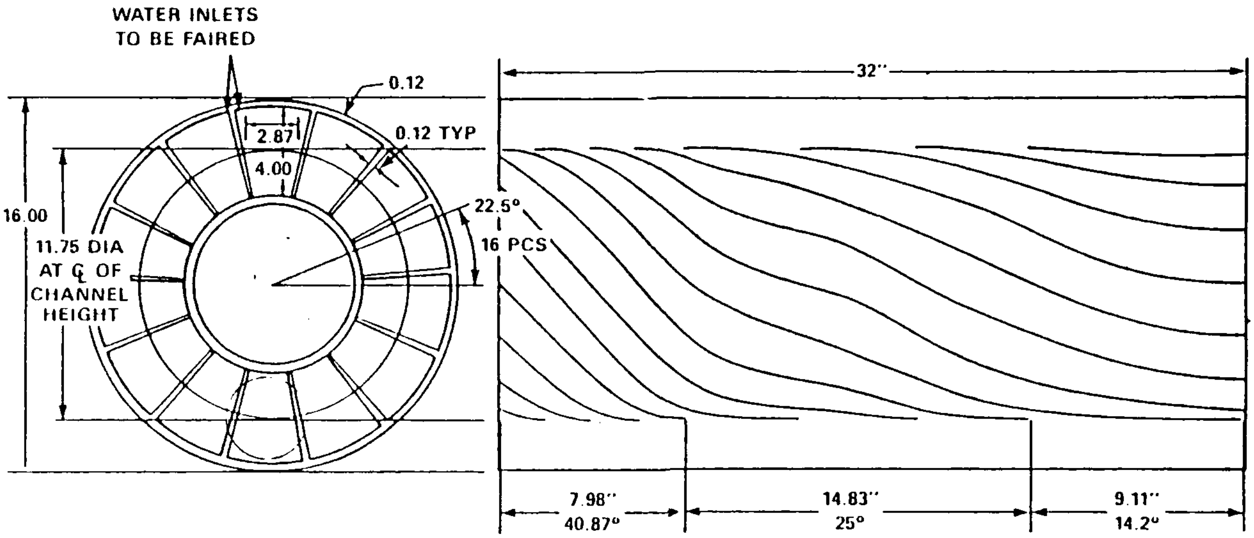


Figure 2. Water Piston Propulsor Rotor

Channel Height: 4 inches
Rotational Support: Central Shaft with Thrust Bearing

3.1.8.3 Seal Geometry and Effectiveness

It was assumed that gas injection would accomplished through a face seal (Fig. 1 B&C). A total arc of 90 degrees is located behind the seal with a single gas port near the leading edge. The seal covers the 90 degree arc between a 4-inch radius and an 8-inch-radius. This configuration is shown in Figure 1A. The gas injection port is circular with a radius of 1.4 inches and line with the combustor. The combustor is anchored to the backplate. It was anticipated that one surface on the backplate would form the seal between adjacent rotor channels. If the gas port dimension is different from the combustor diameter, a transition piece would be provided to accommodate the dimensional change. The seal and backplate were exposed to pressure within the blocked channels of 225 psia at the first channel and a pressure of approximately 60 psia at the last channel. Under the influence of these pressures, the face seal leaks no more than 10% of the inlet flow (i.e., 0.11 lbs/sec). It was assumed that the face seal would utilize some form of ablative seal where a close tolerance is maintained through wear processes. It was thought that materials such as those used to seal gas turbine blades with their housings may be applicable.

Alternately, gas injection could have been accomplished through the core of the rotor. This method has the advantage of gas injection downstream of the seal. With downstream gas injection, the seal area remains flooded and the higher viscosity of the

liquid contributes to the sealing action. However, it was difficult to achieve sufficient injection area with this concept. Furthermore, the sliding seal surface is not easily cooled in this configuration. Therefore, the procurement specified a face seal unless the offeror can present superior technical arguments for an internal inlet.

3.1.9 Combustion System Life

The life of a Marine Corps amphibious vehicle was assumed to be twelve years with 100 operating hours per year. Of this total operating time, it may be assumed that 240 hours are spent in-water. A major rebuild of the combustion system is acceptable after six years. Using a factor of safety of 1.5 in the above arguments, the combustor system should be designed for an operating life of 180 hours over a six year period. Although actual periods of operation may vary from 10 min. to 3 hours, a typical operating period - may be one hour in length. It was anticipated that corrosion may be the limiting factor in combustion system life. It should be noted that amphibious vehicles may be exposed to salt spray over a significant portion of their non-operating life. Fresh water is not to be carried onboard for system waterwash during the mission.

3.1.10 Rotating Drag And Speed

The helix angle at the exit of the rotor determined the rotational torque that is available to overcome *the hydrodynamic drag*, the bearing losses and the seal friction. Since the exit angle is generally fixed, it is important that the drag remain relatively constant. It was estimated that the seal drag should consume no more than one horsepower at full rotational speed (i.e., 9.5 lb-ft @ 550 rpm).

3.1.11 Combustion System Efficiency

The combustor is capable of operating at 95 percent efficiency at full load. Efficiency is defined as the ratio of energy liberated in the combustor to the potential chemical energy contained in the fuel supply. Furthermore, the pressure drop through the combustor shall be no more than 7 percent.

3.2 COMBUSTION SYSTEM DESIGN AND DESCRIPTION

3.2.1 Combustor

Requirements - The prime requirements for the combustion system for the water piston propulsor are an outlet temperature of 3000°F at all operating conditions and an overall pressure drop of 7%. Table 1 shows the estimated operating conditions assuming an axial compressor with no variable geometry. One indication of required operating range is turndown ratio; one example of which is the ratio of maximum temperature rise to minimum temperature rise. This ratio for this application was 1.2, which is modest

compared to gas turbine requirements. Another turndown ratio of interest is the ratio of maximum fuel flow rate to the minimum which is estimated to be 4. This turndown will be discussed in Section 1.4. Combustor temperature rises in the range of 2200°F to 2600°F require that about 70% of the total air be used as primary air which leaves only 30% for dilution and cooling. The outlet temperature of 3000°F imposes severe cooling problems. The cooling design is discussed in later in this section.

Table 1
Estimated Combustor Operating Parameters with Axial Compressor

Speed (%)	Airflow		Pressure (psia)	Inlet Temperature (°F)	Temperature Rise (°F)	PHI	Fuel Flow (Lb/Hr)
	Total (Lb/Sec)	Primary (Lb/Sec)					
100	1.1	0.76	225	800	2200	0.85	162
92	0.98	0.68	200	762	2238	0.855	144
75	0.73	0.51	150	657	2343	0.89	112
60	0.49	0.34	100	522	2478	0.94	79
37	0.24	0.17	50	323	2677	1.00	41

Aero Design - The aerodynamic design of the combustor features a swirl stabilized primary zone with all the primary air entering through the swirler (see Fig. 3). The swirl stabilized approach provides a strong recirculation of the hot products of combustion into the incoming air and fuel which provides a source of ignition. The recirculation also enhances complete burnout of the fuel which minimizes smoke and carbon particulate production. The low particulate emissions are particularly desirable for this application to avoid fouling of the rotor surfaces and the outer liner.

As will be discussed later impingement cooling was used. The pressure drop across the impingement sleeve is 2.5%. Allowing 1% for the air delivery, the remaining 3.5% of the allotted 7% overall was assigned to the swirler and the film cooling air holes. Experience indicates that 3.5% pressure drop is more than adequate to produce the desired degree of recirculation. The swirler is axial to minimize separation on the inner wall and consists of 14 blades set at an angle of 45 degrees.

Mechanical Design - Referring to Figure 3 the combustor consists of an outer liner, an impingement sleeve and the swirler assembly which includes the fuel injector. The outer liner assembly includes the dome, the outer liner, one film air cooling strip and the exit seal. The material chosen was RA333 with a SN-5A coating. The material selection is discussed in a later section. The impingement sleeve causes the inlet air to impinge the outer liner, cooling it, and preheating the air to the swirler. The swirler assembly consists of swirler blades which are 0.020 inch thick and are brazed to the centerbody. The swirler blades are a slip fit into the dome piece. This allows the removal of the outer liner from the rear for inspection.

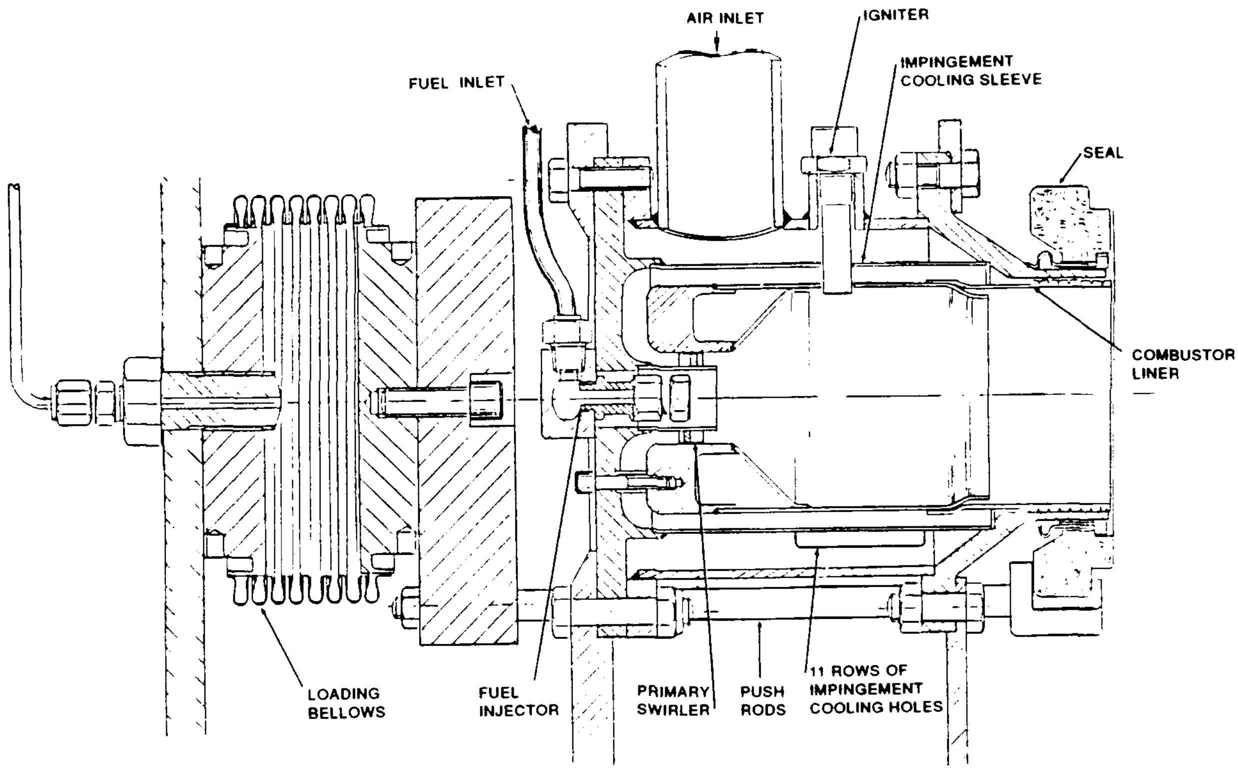


Figure 3. Cross-Section of the WPP

Fuel Injector - The fuel injector design was an integration of the fuel injector and the combustor swirler to produce a form of air blast system. The fuel enters the injector through a 0.045 inch diameter tangential hole, which produces a swirling flow in the injector body. This swirl is preserved through the tube and final orifice. The fuel leaves the final orifice in a wide angle spray with an angle of 110 degrees. This angle is larger than the air swirl angle to enhance mixing of the air and fuel. The fuel is well atomized at this point but is further atomized when it meets the swirling primary air. The type of fuel injection was chosen to provide the required atomization and because it has a minimum number of passages which can plug during shutdown. The fuel flow turndown ratio of 4 will result in an injector pressure drop turndown of 16. In order to minimize the maximum fuel pressure required, the minimum pressure drop must be 20-25 psi. The type of injector chosen demonstrated satisfactory performance at these pressure levels and fuel flow rates. If the minimum pressure is 20 psid then the maximum fuel injector drop will be 320 psid. Therefore, the maximum fuel control outlet pressure will be 530 psig (320 + 210). This pressure level is well within the capability of the fuel control.

Cooling Analysis and Design - The combustor liner cooling design was based on the aerothermal design criteria defined in the RFP, Section 3.1. The design criteria used for the liner cooling was as follows:

Combustor Inlet Temperature:	800°F
Combustor Average Exit Temperature:	3000°F
Compressor Exit Pressure:	225 psia
Overall Allowable Pressure Loss:	7%
Combustor Inlet Pipe Pressure Loss:	1%
Cooling Passage Pressure Loss:	2.5%
Combustor (Swirler) Pressure Loss:	3.5%
Combustor and Liner Cooling Flow Rate:	1.1 lbm/sec
Cooling (Dilution) Flow Rate:	0.4 lbm/sec
Primary Zone Fuel Air Ratio:	0.064
Primary Zone Flame Temperature:	3860°F

The initial cooling analysis of the combustor was performed on a totally film cooled liner. The analysis was run using a Solar program for film cooling (P524) which was developed on Solar's Mars engine design program. This program was developed based on a liner combustion side heat transfer correlation developed by A. H. Lefebvre and M. V. Herbert (Ref. 1) and a film cooling correlation developed at the Lewis Research Center under contract to NAS (Ref. 2). The validity of this program was confirmed on the Mars engine development program and was found to be a useful tool in film cooled combustor liner design.

The results of the film cooling analysis indicates liner hot spot temperatures in excess of 2000°F. This result indicates another type of cooling design is required to achieve acceptable liner metal temperatures.

A regenerative cooling system which uses all the combustion air and cooling air should produce the most effective cooling for a combustor liner with high combustion heat transfer loads. A convective impingement cooling system was chosen for the combustor to provide maximum primary zone cooling heat transfer coefficients. This design also included a film cooling strip in the secondary zone for additional cooling and dilution. The aft end of the combustor is cooled with an annular channel with diamond shaped pin fins to provide a controlled air gap and augment convective cooling of the liner exit. The impingement cooling configuration provides high heat transfer coefficients with the use of the entire combustion air supply (1.1 lbm/sec).

The impingement cooling design of the liner consists of an outer liner with 11 rows of impingement holes of different diameters (Fig. 4). The Kercher and Tabakoff correlation (Ref. 3) was used for the impingement heat transfer analysis. The impingement cooling was designed to produce the maximum heat transfer coefficients in the primary zone (upstream of the film strip) where the combustion heat load is the greatest. Table A-I of the appendix presents a sample output of the impingement program used for the primary zone cooling analysis. The program determines flow rates of the individual rows of impingement holes and calculates the resultant average heat transfer coefficient for each row of holes.

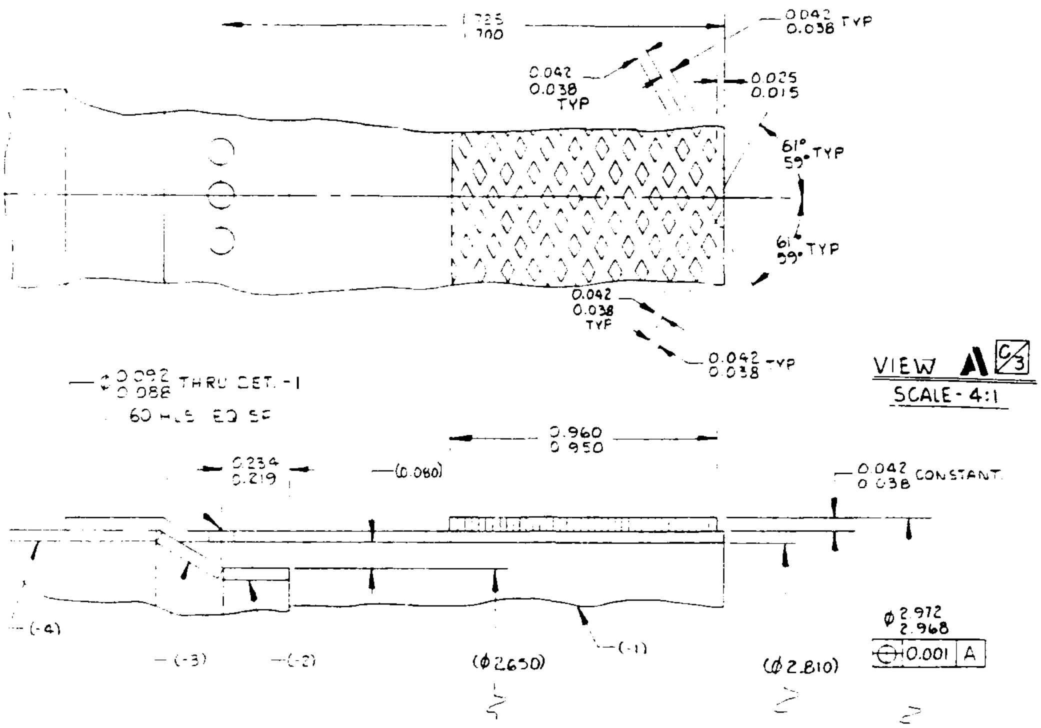


Figure 4. Combustor Aft Cooling

A single film strip was used near the exit of the combustor (see Fig. 3) to augment the impingement cooling and provide secondary zone dilution. The heat transfer correlation used for the film cooling analysis is presented in Reference 2.

The last one inch of combustion liner was cooled with diamond shaped pin fins in an annular gap between the liner and the combustor case (Fig. 4). The pin fins produce augmented convective cooling of the liner and produce a tightly controlled gap between the hot combustion liner and the combustor case. This passage allows a slip fit between the hot liner and cooler exterior combustor case to allow for the mismatch in thermal growth. The results of the thermal analysis of the combustor liner indicated that the maximum liner temperature would not exceed 1650°F based on a maximum predicted heat flux of 475,000 Btu/hr ft2.

3.2.2 Ignition System

The proposed ignition system was a 24 vdc low tension capacitor discharge system. Low tension means that the discharge voltage is 5000 volts maximum. It is commonly used in gas turbine engines. The other commonly used system is a high tension (discharge voltages up to 20,000 volts) capacitor system. The major differences are the discharge voltage and the ignitor plug design. The low tension system was chosen for this application because the combustor and ignitor plug will be damp when lightoff is attempted. The low tension system will function in these conditions while the high tension

ignitor plug tends to short out when wet or damp. Low tension systems are used when the start fuel spray is likely to strike the ignitor because low tension ignitor plugs will spark even when immersed in liquid. The reasons will be discussed below under ignitor plug description.

Exciter - The exciter is a hermetically sealed device which converts a dc input voltage into 2500 to 3000 volt pulses. It includes a circuit which converts the input dc voltage (10-24 vdc) to approximately 3000 vdc by converting the voltage to ac, stepping up the voltage, then rectifying it back to dc. This voltage charges a storage capacitor until a preset level is reached. When this level is achieved the capacitor discharges across a spark gap in the exciter. Exciters are typically intermittent duty devices used only during the lightoff sequence. The performance of an exciter is specified in terms of the energy stored in the capacitor (usually expressed in joules) and the spark rate (sparks/second). Typical values for this type of exciter used on the Solar Titan gas turbine are 0.8 joule and 10 sparks/second. These values are considered to be more than satisfactory for the proposed combustor design.

Ignitor - The ignitor plug used in a low tension ignition system is called a shunted surface plug. An example is shown Figure 5, Type 4. Types 1 through 3 are examples of high tension plugs. There is a semi-conductor material between the center and outer or ground electrodes. The purpose for this material is to provide an initial path for the exciter discharge to travel which ionizes the air in the gap between the electrodes allowing the main portion of the discharge to flow. A much lower discharge voltage is required to ionize a shunted surface gap than an air gap which is used with a high tension system. If there is moisture or fuel in an air gap the system is shorted and will not function. A shunted surface type plug will function even when immersed in liquid fuel or water and, therefore, is the ideal choice for this application.

Cable - Ignition cables used on gas turbine ignition systems consist of a center conductor and an outer metal sheathing which serves as the ground return. The center conductor for a low tension system can be smaller gage than a high tension system because it carries lower voltage (3000 vs. 15000). On the ends are special high voltage connectors which are waterproof. The cable for this application incorporates a waterproofing material between the center Conductor and the outer sheath. The outer sheath is made of corrosion resistant material. The ignitor plug end of the cable is under water when the WPP is operating while the exciter end terminates at the exciter in an enclosure mounted on the transom.

3.3 SEAL SYSTEM DESIGN AND DESCRIPTION

The efficiency of the WPP concept is highly dependent upon the effectiveness of the seal, both in preventing gas leakage and in minimizing the rotational torque loss. Because of fixed geometry of the rotor exit, seal drag must be relatively constant over hundreds of hours of use. The seal has been specified to meet the following criteria:

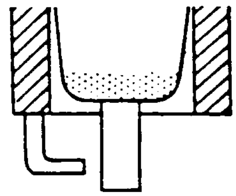
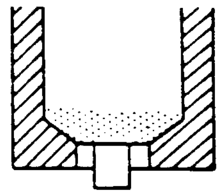
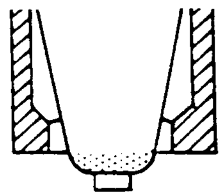
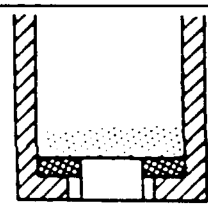
TYPE	TITLE	TYPICAL GAP CONFIGURATION
1	IGNITER PLUG HIGH VOLTAGE AIR GAP	
2	IGNITER PLUG HIGH VOLTAGE SURFACE GAP	
3	IGNITER PLUG HIGH VOLTAGE AIR SURFACE GAP	
4	IGNITER PLUG LOW VOLTAGE SHUNTED SURFACE GAP	

Figure 5. Schematic, Igniter Plugs

Seal Thermal Shock - The seal system will be in contact with 225 psi air at 3000°F at one segment while concurrently being exposed to 40°F water at another segment. Similarly, a single piece of seal will alternately be exposed to 3000°F gas and 40°F. The seal system must resist cracking, fracture, chipping, accelerated corrosion, and accelerated wear under this rapid thermal cycling in a corrosive medium.

Seal leakage shall be no more than 0.1 lb/sec for an operating life of 180 hours over a six year period, given maximum combustor exit pressure of 225 psia.

3.3.1 Seal Design

Two alternate face seal concepts were considered in the project design task, the first being abradable gas turbine tip seals and the second a chrome carbide to carbon-graphite seal.

An abradable turbine tip seal must operate with a net clearance, however small, since these seal systems have a high coefficient of friction and high wear rate when they are in loaded contact. They therefore must maintain a net clearance during the vast majority of operation. This imposes a net leakage past the seal and with the proposed seal system results in localized temperatures on the order of 2700°F at the outer and inner peripheries of the rotor annulus.

The second seal system consisting of a low friction contacting carbon-graphite seal face with chrome carbide rotor face offers the advantage of near zero leakage and low seal face temperature. The absence of seal leakage results in very low surface heat transfer from the 3000°F combustor exit gases because of the low velocities (only on the order of 150 ft/sec). This allows the design of a cool seal face utilizing forced or boiling convective heat transfer on the surface behind the seal face. In the worst case, with the craft stationary at full combustor output, design calculations show that the maximum carbon-graphite face seal temperatures will not exceed 550°F. Carbon-graphite materials have very good durability at this temperature.

Because of thermal shock considerations and better sealing inherent to the contact type seal, the carbon-graphite to chrome carbide face seal system was chosen for the WPP. With this design all seal surfaces are kept below 350°F during operation via backside convective water cooling. This combined with the very low coefficient of expansion typical to carbon-graphite minimizes thermal stress and thermal shock effects. Thermal distortions are also minimized as is shown below. Better sealing is achieved due to contact and wear in seal to rotor surfaces. The low friction coefficient of carbon graphite proves an advantage in maintaining consistently low seal drag forces.

Seal wear was expected to be minimal over the seal duty cycle. Reference 4 cites experimental seal wear rates of resin impregnated carbon sealing warm water at velocities and pressures similar to the WPP application. The maximum wear rate in any of the tests was 0.005 inch per 100 hours.

The seal material selected for the WPP application was P-658RC. This is a resin impregnated graphite carbon material manufactured by Pure Carbon Company. The selection was based primarily on sea water resistance of this material along with other desirable properties such as a low coefficient of thermal expansion, good wear properties and good strength. An alternate material, P-658RCH, was considered as a candidate for spares/replacement because of slightly higher temperature capabilities. In the event that forced convective cooling were lost and boiling convective heat transfer existed at the back face, this capability to 600°F may be necessary.

Table 2 lists properties of the P-658RC and P-658RCH graphite carbon seal materials.

Table 2
Physical Properties of Two Resin-Filled Carbon Graphite Seal Materials (Pure Carbon Co.)

Material Property	Material	
	P-658RC	P-658RCH
Strength (psi)		
Compressive	34000	31000
Transverse	11500	9000
Tensile	7000	6500
Modulus of Elasticity (psi x 106)	3.5	3.3
Temperature Limit (°F) Oxidizing atm.	500	600
Coefficient of Thermal Expansion (in.in./°F x 10-6)	2.7	2.6
Thermal Conductivity (Btu/hr/ft2/°F/ft)	5	5

3.3.2 Seal Leakage

Seal leakage requires the rotor inlet face to be in close contact with the seal face. Leakage flow (lb/sec) due to a gap along a single thin rotor vane can be determined from:

$$W=0.532*Cd*A*P/\sqrt{T}$$

Where: Cd is coefficient of discharge taken as 0.65
A is area - inches squared
P is pressure - pounds per square inch absolute
T is temperature - degrees Rankine

Given pressure in the rotor chamber varying continuously from 225 psig at combustor end to 60 psig at the opposite end and dividing into three regions for simplicity, the maximum allowable average gap along the 27 inch periphery for 0.11 lb/sec flow is approximately 0.004 inch.

The approach chosen here for the seal face was to maintain these surfaces at a relatively cool temperature (<350°F) by providing water cooling at the back face. This prevents the possibility of thermal distortion of the seal which could create unacceptable gaps in the seal between rotor and seal sector. Graphite carbon materials have relatively low expansion coefficients and good thermal conductivity which minimize this problem.

However, these materials have a low elastic modulus (about 3 x 106 psi) and would be expected to deform unacceptably under the significant gas pressure loads of the WPP. A high modulus superalloy back-up plate attached via struts to the graphite seal sector has been included in the seal design to solve this difficulty.

Evaluation of seal system deformations due to thermal and mechanical loadings as well as stresses due to these sources was made by finite element ANSYS computer analysis.

Figure 6 shows the 3-D finite element model of the graphite seal and backup plate. The model geometry was varied in design analysis work to provide low seal face deformation and low seal stresses. The model includes the effects of back side water cooling of the graphite seal along with convective heating of the active seal face from 3000°F combustion gases. Gas pressure loads were also applied in the 3-D ANSYS analysis to determine net deformation and stress of the graphite seal sector.

A deflection plot is shown in Figure 7 and stress plot in Figure 8. The maximum deflection is less than 0.004 inch which assumes that leakage will be below 0.1 pps. The maximum stress is 2000 psi. The P-658RC graphite carbon seal has tensile strength of 7.0 ksi allowing a factor of safety of 3.5.

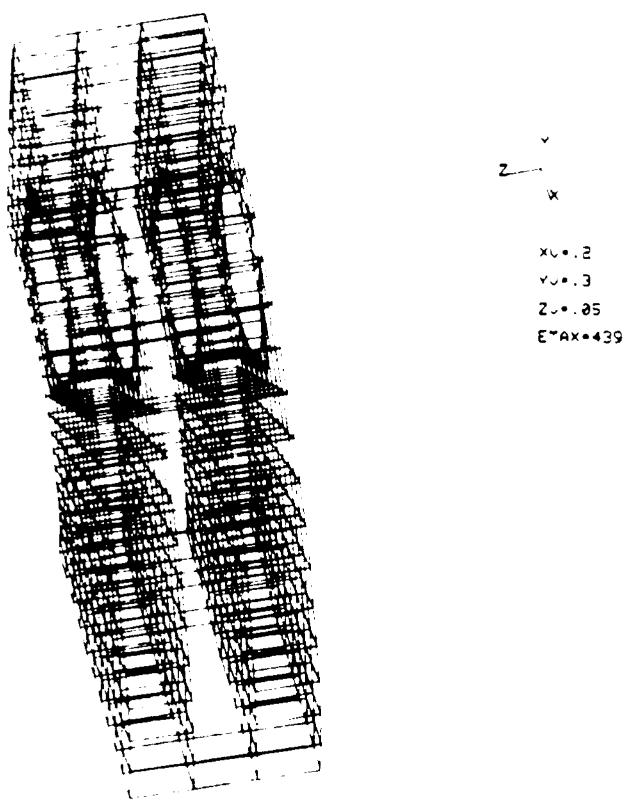


Figure 6. 3-D Finite Element Model of Graphite Face Seal and Back Plate

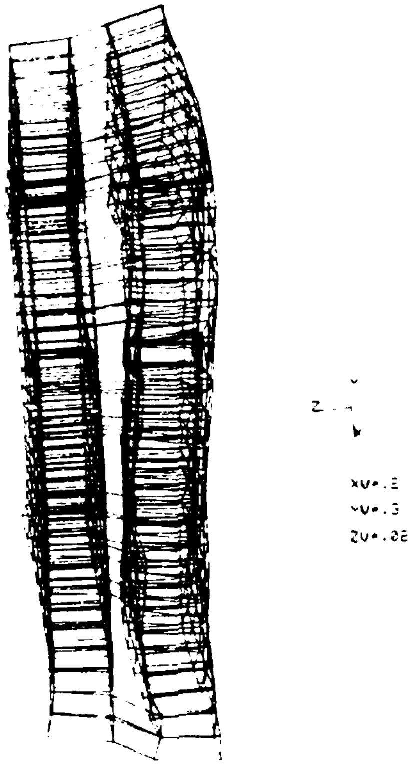


Figure 7. Deflection Plot of Graphite Face Seal and Back Plate

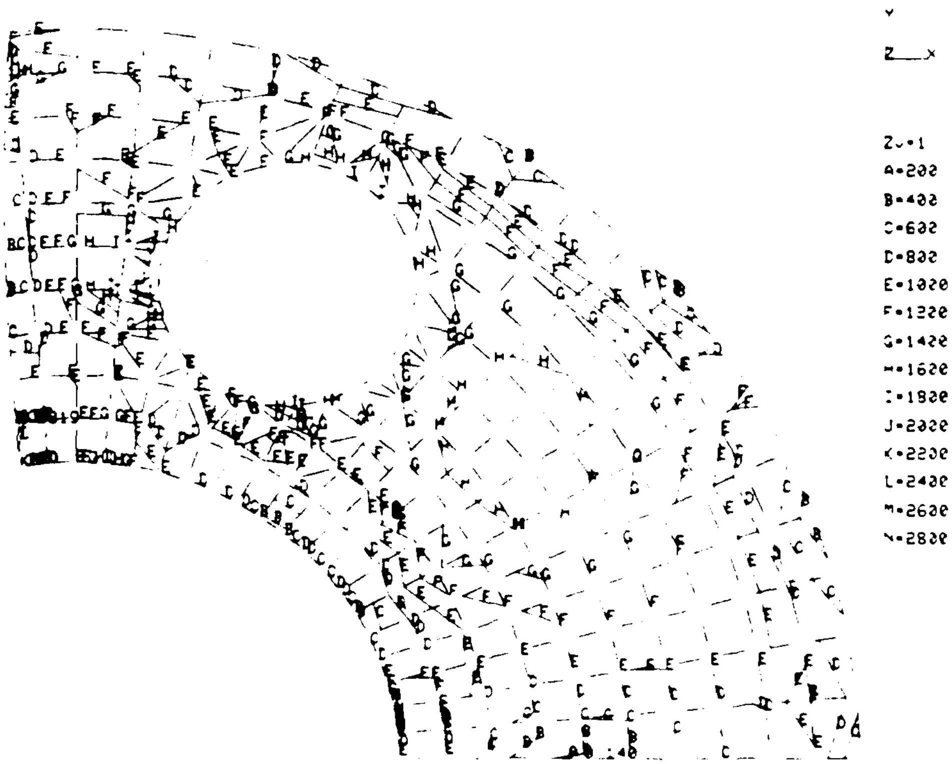


Figure 8. Stress Plot of Graphite Seal Face and Back Plate

The seal design is shown in Figures 9. The seal face will have proper wear characteristics with the rotor mating face fabricated from a hard material with Rockwell C hardness greater than 45. Chrome carbide, titanium carbide or similar hard materials would be suitable from a wear standpoint.

The seal system is loaded against the rotor front face with a two element bellow system. Compressor secondary air is used to pressurize the bellows. The control system uses combustor pressure to determine proper bellows pressure in a closed loop scheme. The bellows size and placement allows them to balance the nominal 4000 lb thrust force due to gas pressure loads in rotor chambers and provide an additional load of up to 165 lb for face seal contact force. The bellows seal loading system was also designed to accommodate rotor angular misalignments of up to +1/4 degree.

3.4 CONTROL SYSTEM DESIGN AND DESCRIPTION

There are two independent control loops within the WPP combustor and seal control system, the combustor or fuel control loop and the seal control loop. For the fuel control system, Solar defined the control strategy, the hardware specification and the development of software. For the seal control system Solar defined the control strategy and the hardware specification, only.

3.4.1 Fuel Control System

The fuel control system is an open loop system shown schematically in Figure 10. utilizing air flow measurement, combustor inlet temperature and an algorithm for temperature rise for diesel fuel. A closed loop system was not considered because of the difficulty of sensing the desired rotor inlet temperature of 3000°F. Thermocouples operating in such an environment have relatively short service time. Also multiple sensors would be required to obtain the true mixed average.

An equation has been developed for temperature rise due to the combustion of Marine diesel fuel as a function of fuel-air ratio and inlet temperature. If the inlet temperature and airflow are known then the fuel flow required to achieve 3000°F outlet temperature can be calculated. This equation is:

$$W_f = Wa*[0.1366 * Ck * (3460\text{-}Tin)/10680 - Ck * (3460\text{-}Tin)]$$
$$C_k = [0.00014 * (Tin - 600)] + 1$$

The combustor inlet temperature is measured with a J-type thermocouple in the combustor swirler area. To measure the airflow, the pressure before and after the impingement cooling sleeve is measured with pressure transducers. The impingement cooling sleeve was calibrated and its effective flow area was determined. Since the inlet temperature used for the flow calculation is different than the inlet temperature at the impingement sleeve, a second J-type thermocouple was used. The pressure and

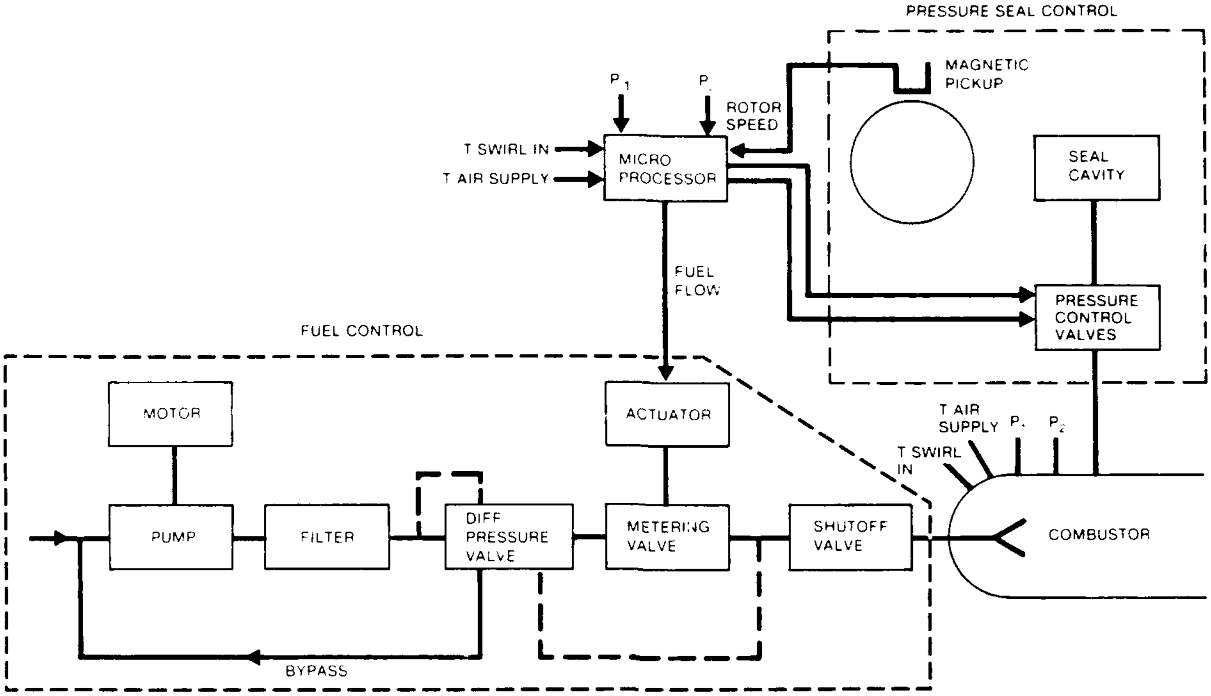


Figure 10. Control System

temperature signals are sent to a microprocessor which calculates the required fuel flow and output a signal proportional to the required fuel flow. The equation for airflow is:

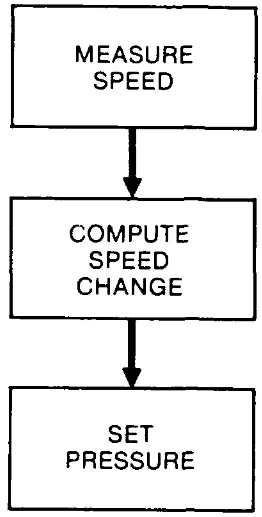
$$W = K\,[PI\ (PI - P2)/T]_{1/2}$$

where K is determined by the calibration test.

The fuel flow output signal is sent to a gas turbine electronic fuel control assembly. The control assembly includes a gear type fuel pump, a relief valve, servo valve, shutoff valve and delta pressure valve. The assembly is shown schematically in the box labelled fuel control in Figure 10. The valve accepts a 0-100 ma signal and varies its area in direct proportion to the input signal. The differential pressure valve (ΔP) maintains a constant pressure drop across the servo valve by bypassing excess flow back to the pump inlet. By virtue of the constant servo valve pressure drop, the servo valve opening is directly proportional to flow and therefore fuel flow is directly proportional to input signal.

3.4.2 Pressure Seal Control System

Solar defined two control strategies for the pressure seal control system. Both strategies are closed loop and require only one or two microprocessor inputs. The first strategy maximizes rotor speed for any given combustor outlet condition and are shown in Figures. 11A and 11B. In this scenario the seal plate pressure loading oscillates between a small



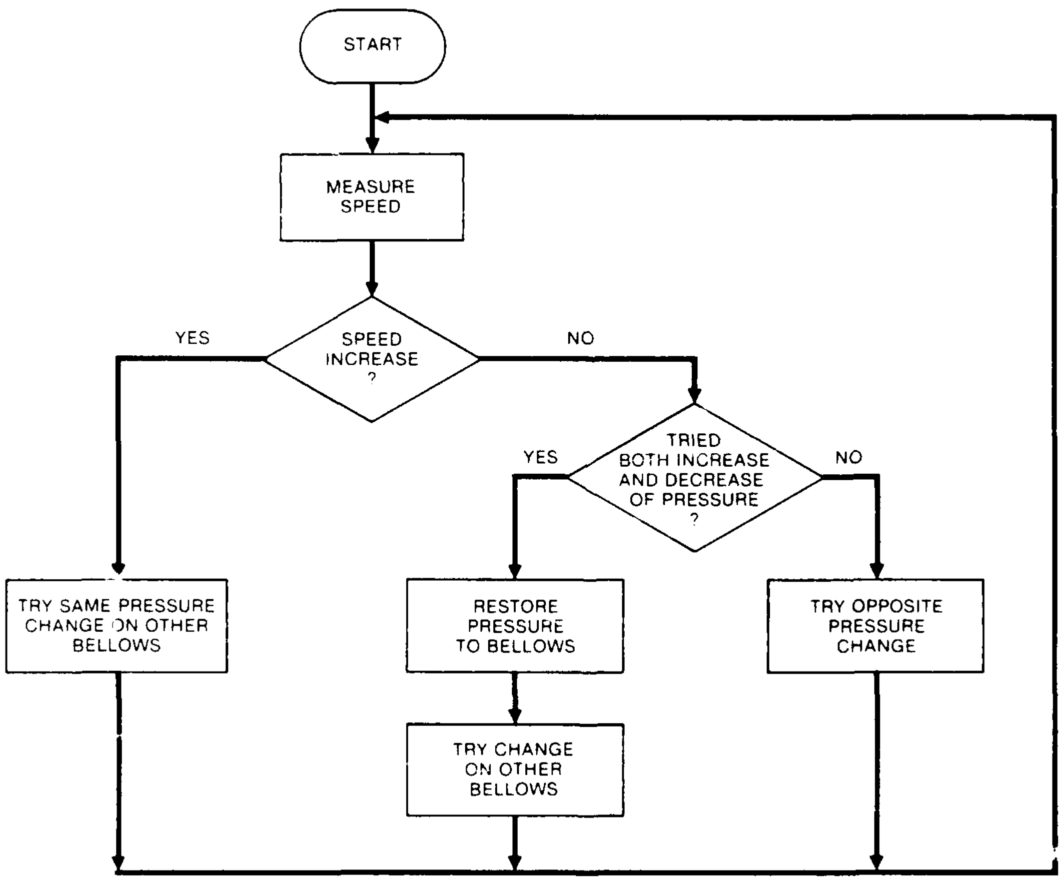


Figure 11. Simplified Pressure Seal Control System Flow and Pressure Seal Control System Flow Chart

leakage gap and a small seal-rotor friction load. In this manner a maximum rotor speed can be insured. Rotor speed is the only microprocessor input required of the strategy. The one drawback of this approach is that the pressure seal must come out of contact with the rotor. When this condition occurs, it is likely that seal plate erosion rates will increase. This possible drawback is eliminated in the other of the two possible control techniques. In the second strategy a unique rotor speed would be specified for any given combustor airflow. That speed would correspond to a seal friction loss less than one horsepower, but would provide a positive face seal load. This rotor speed-combustor airflow relationship would best be determined experimentally. The pressure bellows would then be loaded until this predetermined speed specification was obtained. The erosion problems possibly encountered due to exhaust gas blowby would be eliminated since the seal would never come out of contact with the rotor. The flowchart for this strategy can be seen in Figure 12.

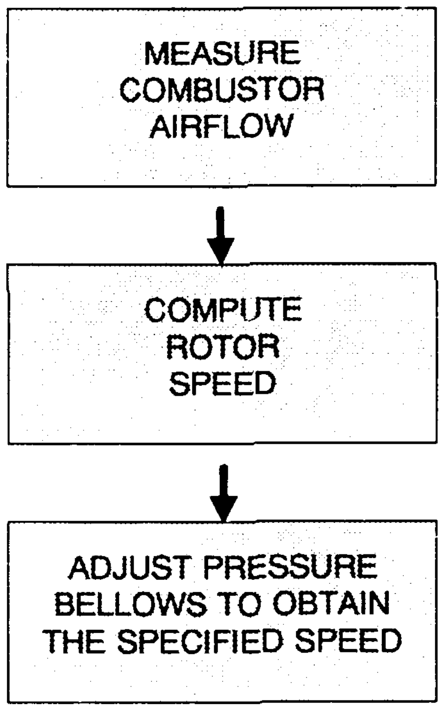


Figure 12. Seal Control System

The control hardware required of the seal system, excluding the microprocessor, consists of a rotor speed sensor and I-P pressure regulators. The rotor speed will be found with a magnetic pickup that reads timing marks placed on the outer diameter of the rotor. The I-P controllers will regulate the pressures within the seal bellows system as specified by the system software.

The two control systems required of the WPP combustion and seal system are processed with an Analog Devices μMAC5000 control and data acquisition system. This device, in addition to its control tasks, allows real-time data acquisition and monitoring when interfaced with a microcomputer such as an IBM-PC. This capability will facilitate the control system tune-in as well as the documentation of test runs. In addition, the μMAC5000 is fully expandable which allows additional system inputs and/or outputs. This could prove advantageous in the event that input signal checking or averaging be required or if other parameters need be determined.

3.5 MATERIALS SELECTION

Solar chose to select an all-superalloy system based on the pitting tendency of stainless steel (up to 0.113 inch deep in 3 years, Ref. 5) in quiet, ambient sea water and the tendency for stainless steel to stress corrosion crack at temperatures above 140°F (Ref. 6). Exceptions to this are the bolts which are exposed to temperatures less than 140°F and the support bars and clamps used to hold the graphite wear plate.

The combustor liner is fabricated from RA333 and coated with a vitreous enamel coating based on the Solar hot corrosion data. Flooding of the combustor at relatively frequent intervals results in a residue of sea salt on the surface of the combustor alloy. This residue can result in extremely high rates of hot corrosion. Data was developed at Solar on a number of sheet metal alloys. In this test, specimens were coated with 1 mg/cm of $95Na_2S0_4$-5NaCl and then exposed at various temperatures in 15 hour cycles. The specimens were weighed before and after each cycle. The time to zero failure, i.e., a loss of 200 mg/cm2 was thus determined. The results are shown in Figure 13. As can be seen, few alloys can resist this corrosive media for more than 200 hours at the typical combustor wall temperature of 1600°F. The most resistant alloy, RA333, is the one that was selected for use in the combustor fabrication. At I500°F the life of the alloy under these severe conditions is 250 hours compared with 125 hours for Hastelloy X and 200 hours of for Haynes Alloy 188. This same data is also presented in Figure 14 for the maximum penetration per hour for the same alloys as determined metallographically. These results also confirm the superior performance of RA333 to the other alloys listed. Based on these results, Solar maintains that the material chosen can meet the life goals of the combustor.

The outer combustor liner was fabricated from Hastelloy X sheet. This is the current material used on all Solar gas turbine combustor liners. The lower temperature encountered in the outer liner precludes the necessity for a coating on a more corrosion-resistant material. The outer combustor case and pre-heated air inlet were fabricated from Inconel 600 pipe based on availability and the materials resistance to chloride stress corrosion cracking. The use of pipe facilitated manufacturing and reduced unnecessary machining for these components.

Hastelloy C-276 was selected for the nozzle plate primarily for its availability and resistance to sea water corrosion (Ref. 7). The cooler temperatures and shielding from the sulfur-containing exhaust gases allow this material to be used without a coating.

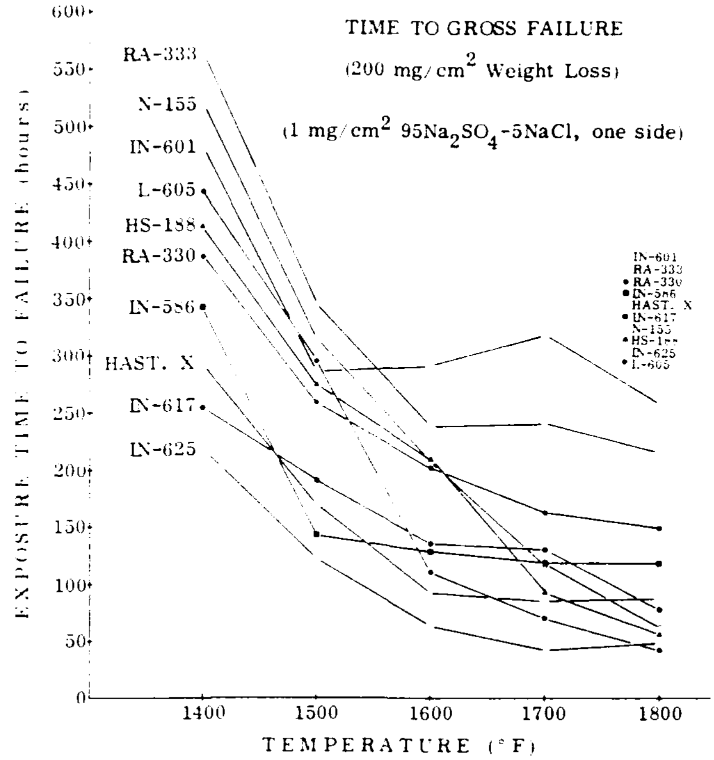


Figure 13. Hot Corrosion of Uncoated Sheet Alloys in Slow Moving Air

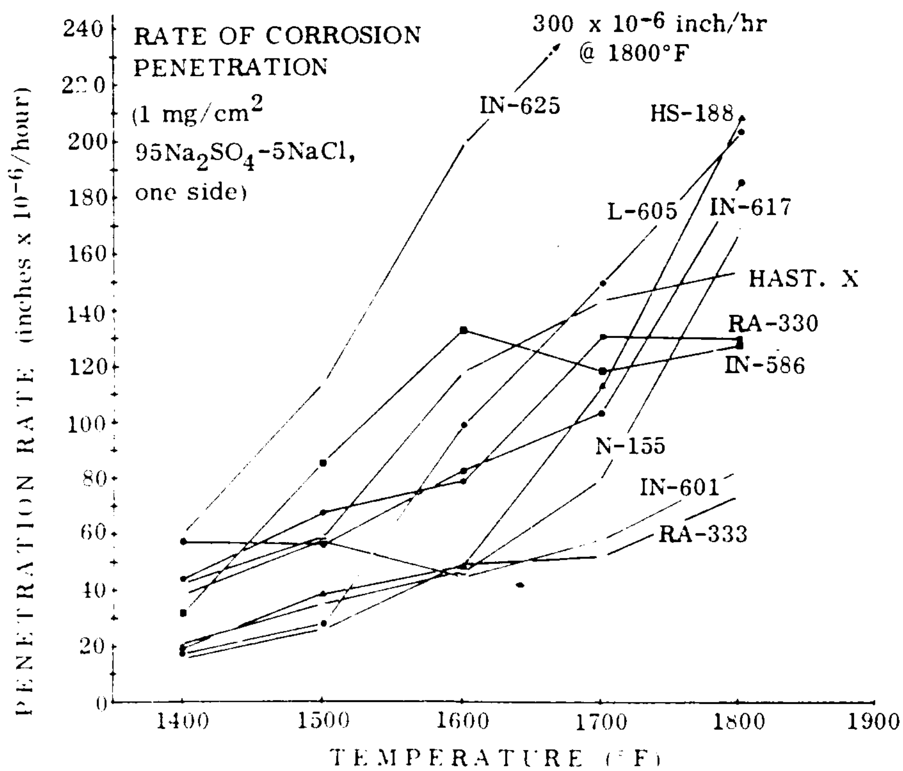


Figure 14. Penetration and Loss in Thicknesses of Wrought Alloys Per Hour

The choice of an all superalloy system prevents the occurrence of any galvanic corrosion when in contact with dissimilar materials. The only exception is for such standard parts as bolts, rods, and ignitor which are used on Turbomach (formerly Solar) Titan engines. The 347 CRES ignitor housing, directly exposed to the combustor exhaust, is protected with a vitreous enamel coating. Stress corrosion cracks do not impair the ignitor's function since this is not a load-carrying member. Inconel X-750 bolts are used to attach the combustor liner base to the base of the combustor can.

RA333 bar stock is used for the combustor liner base to match to the thermal expansion properties of the RA333 inner liner. The remainder of the combustor case, including the base and support plate, is fabricated from Hastelloy C-276 based on the materials outstanding resistance to seawater corrosion and availability in plate stock. Hastelloy X was selected for the fuel line tubing and support plate bellows based on a much lower cost, availability and good corrosion resistance in seawater.

To prevent galvanic corrosion of 316 CRES bolts in contact with Hastelloy C276, the bolts are painted with an epoxy amine primer and installed wet. For those areas experiencing warmer service temperature such as perhaps the contact areas between 316 CRES support and graphite wear plate, a vitreous enamel coating was applied to provide insulation from a possible galvanic couple.

Type 316 CRES was chosen for bolt material for pitting resistance. Stress corrosion cracking should not be a problem for applications where the service temperature is less than 140°F (Ref. 8). Fittings are to be made from Inconel 600 due to availability and proven stress corrosion resistance (Ref. 9).

All of the materials selected are readily welded by conventional welding procedures. The use of Hastelloy W filler is recommended for dissimilar alloy welds.

4.0

COMBUSTOR RIG TESTS

4.1 OBJECTIVE

The objective of the combustor rig tests was to verify proper operation of the combustor and fuel control system. Proper operation included demonstration of light-off, stable operation over a range of inlet temperature and pressure, and a constant outlet temperature of 3000°F.

4.2 PROCEDURE

The combustor and seal system were installed in the test cell with a stationary plate to simulate the rotor and a back pressure valve to provide the desired back pressure. The air flow was supplied by the experimental test air supply system which has the capability to supply air at the flow rates, pressures and temperature required for the WPP program. The WPP fuel control system was used to control the fuel flow rate as a function of airflow and combustor inlet temperature. The main function of the fuel control system is to supply the correct rate of fuel to maintain the outlet temperature at 3000°F. Verification of the outlet temperature was accomplished by measuring the exhaust gas emissions and calculating the temperature from the emissions and the fuel properties. Pressure drops, air flow and fuel flows were also measured.

4.3 RESULTS

The initial testing concentrated on checking out the rig and systems. The air flow calculated by the microprocessor was checked against that measured by the test cell orifice plate. as discussed above the microprocessor incorporates a simplified orifice equation. The constant in the equation can be changed in the software. The fuel control was calibrated to determine the output flow to electrical input signal relationship which is also in the microprocessor software. Initial attempts to lightoff were unsuccessful and were ultimately solved by a fuel nozzle change. The first tests with combustion were done using a slave electrical signal to the fuel control to allow control of the fuel flow independent of the microprocessor. These tests included verification of the calculated value of air flow by the microprocessor versus an ASME orifice run and of the delivered fuel flow rate versus the calculated value. The next series of four tests were run with the microprocessor in control. In test #1 the microprocessor was programmed for a 2500°F output. Figure 15 shows the outlet temperature calculated from the exhaust composition vs inlet pressure. As can be noted the outlet temperature was not constant. The microprocessor algorithm was changed for the subsequent tests and the programmed outlet temperature increased to 3000°F. As can be seen in Figure 15 the outlet temperature in Test #4 was in range of 3000°F ± 200°. It was felt that a slight change to

the algorithm would reduce the deviation to ±100°. At this point the performance was judged to be satisfactory to proceed with the water channel tests at Tracor.

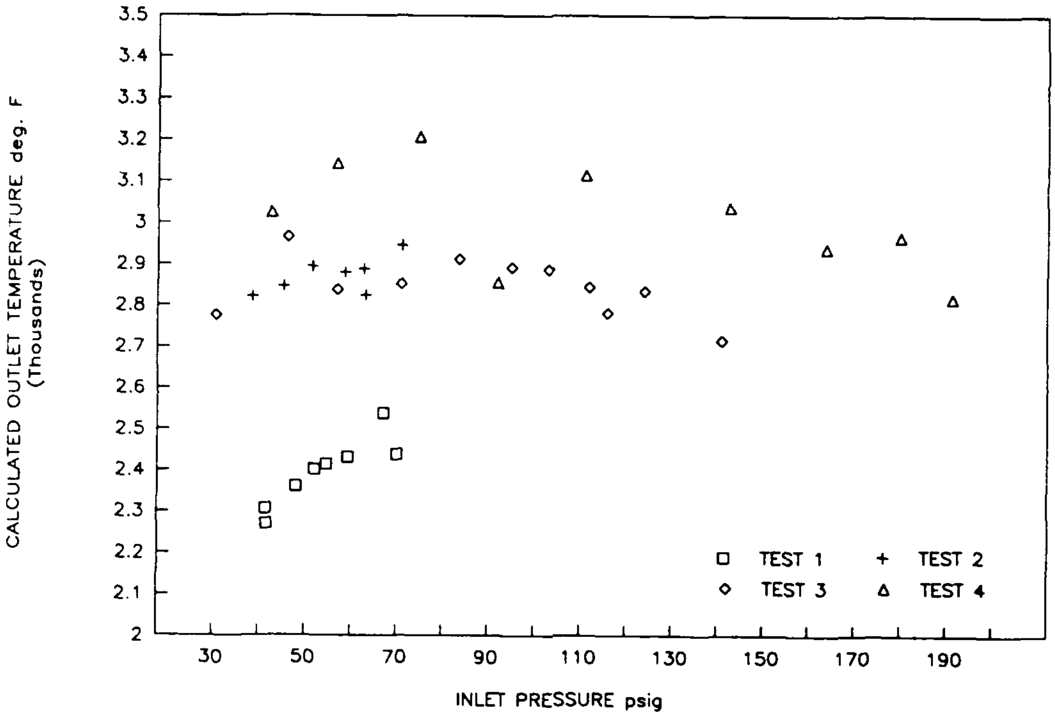


Figure 15. Calculated Outlet Temperature vs. Inlet Pressure During 4, Microprocessor Run Combustor Tests

5.0

SIMULATED ROTOR TESTS

The evaluation of the pneumatically loaded rotor face seal was to be accomplished as an integrated assembly with the combustor in a towing tank or recirculating water channel. In order to minimize expensive test time in a tank or channel, the initial evaluation of the rotor/seal control system was performed on an intermediate rig at Solar before full-scale, prototype tests in water. This rig generated information on seal-to-rotor friction characteristics, rotor/seal wear characteristics, a seal loading schedule, and seal leakage rates.

Initial testing indicated that a P+K regulator system would most easily achieve the desired bellows loading control on the seal/ rotor interface.

5.1 TEST CONFIGURATION

The simulated rotor tests were accomplished using the original combustor assembly together with a 16 inch diameter, 16 channel, 4 inch long, simulated rotor situated at the exit of the combustor. The rotor assembly, shown in Figure 16A and B, was driven by a 5 Hp, variable speed motor, capable of generating the desired 550 rpm at design conditions. A torque meter was used to measure the frictional power loss imposed by the face seal. The rotor and combustor were separated by a seal plate which was attached to the combustor exhaust. Tie-rods, attached to the bellows plate, secured the graphite seal plate by using clips around its perimeter, such that when the bellows expanded the seal was pressed up against the rotor. The bellows and tie-rods are shown in Figure 17. A gas tight seal was created between the seal plate and rotor to prevent combustion exhaust leakage when passing to the rotor. The seal plate with clips are shown in Figure 18.

The rotor was stainless steel with a chrome plated face. A stainless steel back plate, 3/8 inch thick, was welded to the back end of the rotor. This plate had 16, 1-inch diameter holes which were situated near the bottom-center of each channel. With combustor air flowing, these holes simulated the backpressure that would be seen by the prototype when in operation. Separated by a gasket, a 1/4 inch thick slip ring, seen in Figure 19, was mounted to the back of this plate. This slip ring, also having 16, 1-inch diameter holes, could be rotated to open or close off the holes in the back plate, increasing or decreasing the simulated backpressure.

The seal design was based on the face seal concept as previously described. A carbon-graphite seal was chosen. A seal system consisting of a low friction, contacting carbon-graphite seal face with a chrome carbide rotor face offered the advantage of near zero leakage and low seal face temperatures. The absence of seal leakage allows the design of a cool seal face utilizing forced or boiling convective heat transfer on the surface behind the seal face. With the contacting seal design, all seal surfaces are kept below

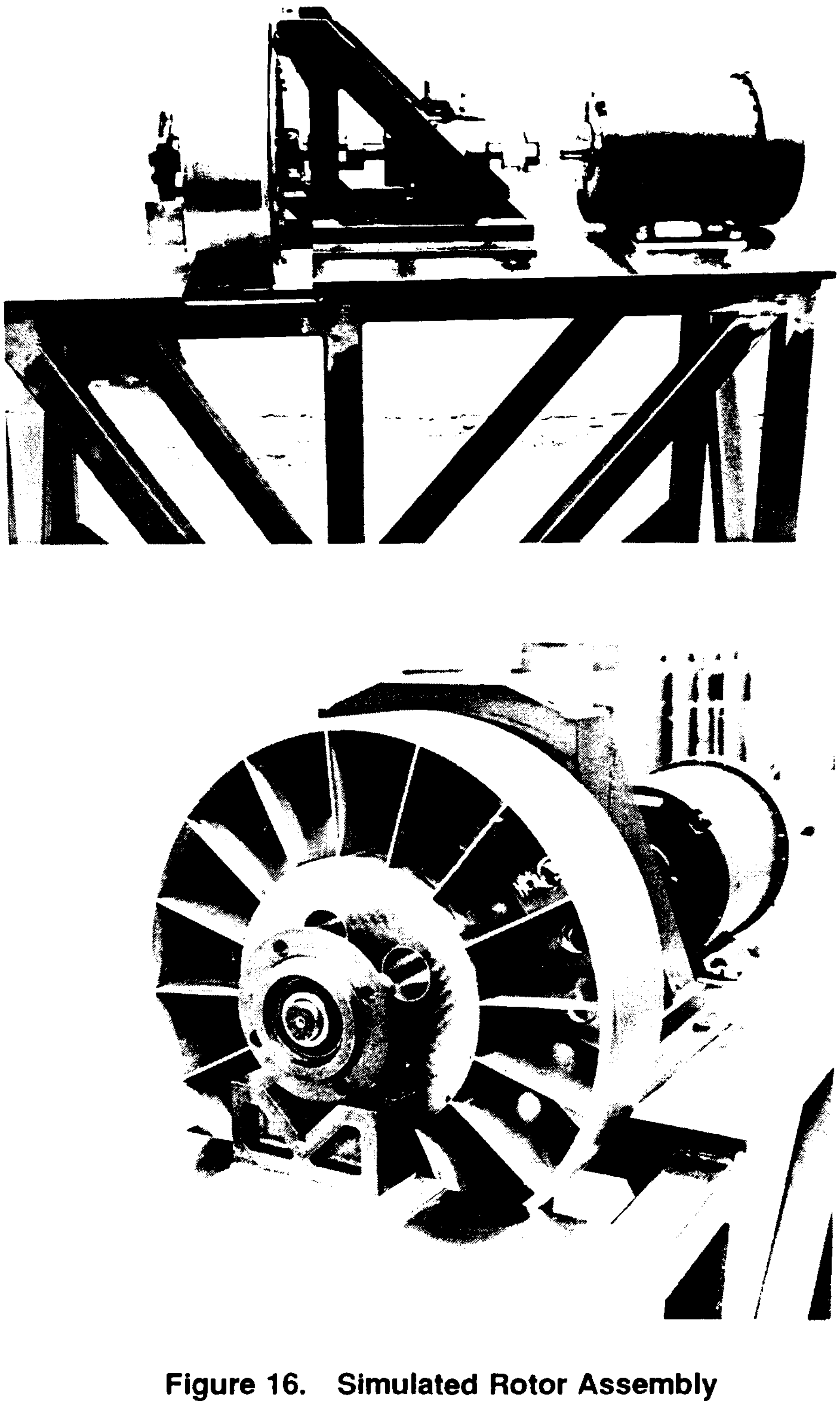


Figure 16. Simulated Rotor Assembly

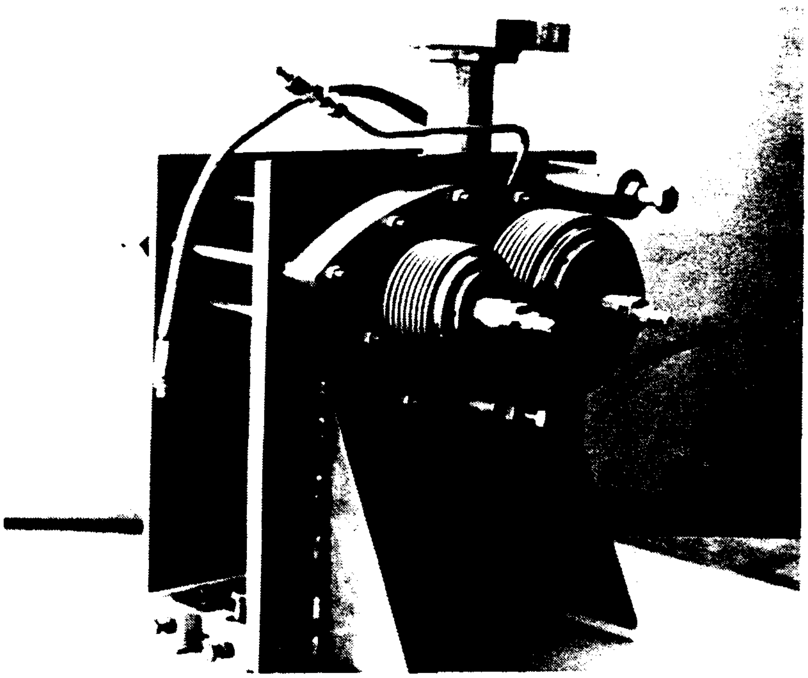


Figure 17. Bellows and Tie-Rods

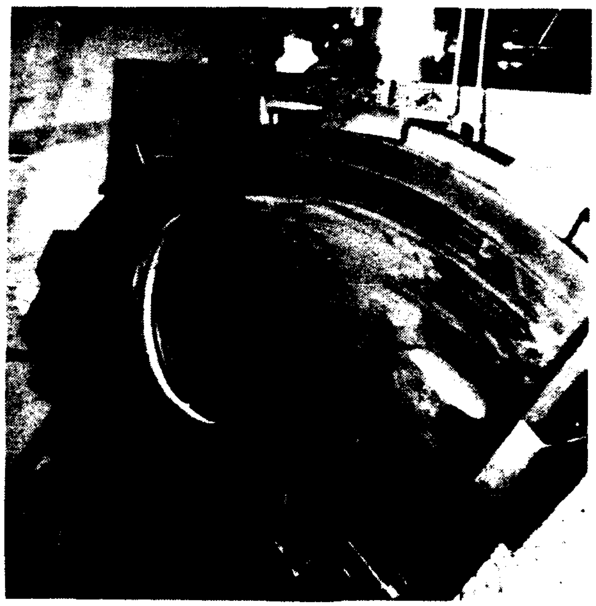


Figure 18. Graphite Seal Plate Being Held in Place by Clips

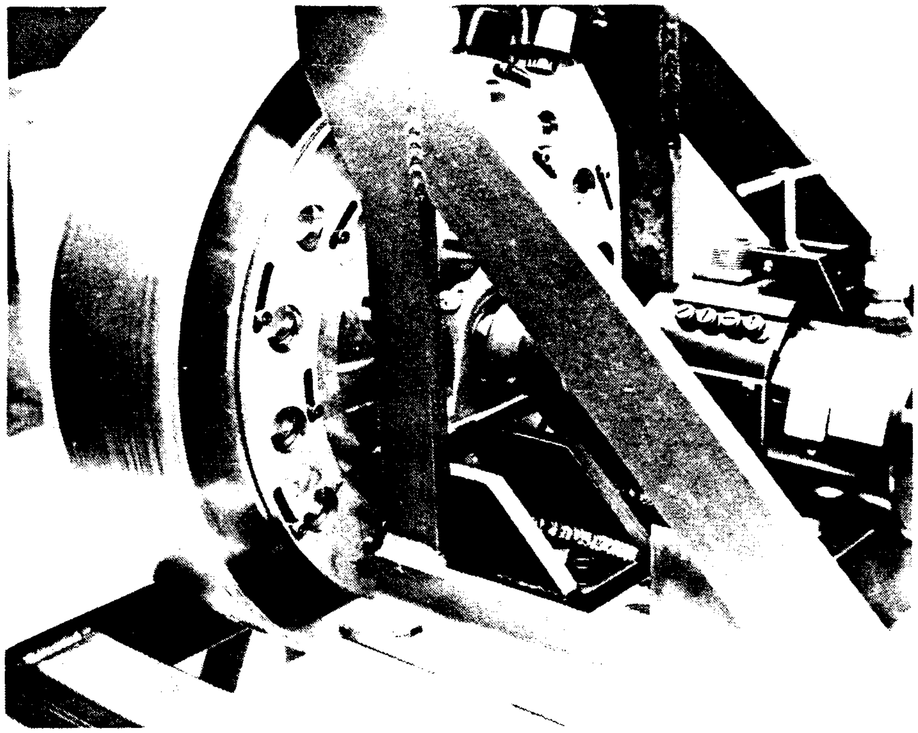


Figure 19. The Back End of the Rotor With Slip Ring

400°F during operation via backside convective water cooling. This combined with a very low coefficient of thermal expansion, typical to carbon-graphite, minimizes thermal stresses and thermal shock effects. The low friction coefficient is another advantage of carbon-graphite in maintaining consistently low seal drag forces.

The seal material selected for the WPP was P-658RC manufactured by Pure Carbon Company. The material is a resin impregnated graphite carbon with operating temperatures of up to 550°F. The material selection was based primarily on its resistance to sea water, along with its low coefficient of thermal expansion, and exceptional wear and strength characteristics.

5.2 RESULTS

Testing began at low combustor pressures and established the rotor to seal gap size. Initially, the seal was backed off the rotor allowing the rotor to turn without the excess drag of the seal. A correlation between the gap size and the amount of bellows pressure required to have the surfaces meet was developed.

The next series of test were performed to establish a bellows loading schedule. Using combustor air, unwanted pressure variables were encountered; therefore further tests were conducted without air. With the rig apart, for realignment purposes, examination of the graphite seal plate revealed pits occurring on the surface of the seal and a vertical

hairline crack through the plate shown in Figure 20. Results of an inspection indicated that pitting had occurred due to the shearing of bubbles that formed on the surface of the plate. These bubbles were created by the frictional heating of a plastic resin compound which was impregnated into the graphite plate. The bubbles formed and with a single pass of a rotor vane, they were sheared off. All subsequent testing was done with a water nozzle spray assembly installed. As a jet of water was sprayed across the seal there was visible evidence of pressurized contact between the rotor face and the graphite seal. This acted to cool and lubricate the plate, and was more representative of testing in water. While maintaining a constant rotor rpm, the bellows pressure loading was varied until a seal was created.

Tests resumed using a combus [illegible] ssure/rotor rpm schedule of 50 psia/137 rpm, 150 psia/210 rpm, and 210 psia/[illegible] m. A bellows control scheme was configured for low combustor pressures. Be [illegible] oading in excess of 60 psig stalled the motor and high combustor pressures were unattainable due to operational constraints of the motor. As a result of these complications, static tests using only combustor air and no rotation were preformed. With this configuration, combustor pressures of 110 psig were achieved and a control scheme yielding a bellows loading of 30-40 psig above combustor pressure was established.

Frictional characteristics of the graphite seal against the rotor were also investigated. Several tests, with and without the water spray, were performed at various combustor pressures and rotor speeds. Calculations revealed a very low coefficient of friction; wet $\mu=0.024$, and dry $\mu=0.050$. During all testing, loading and torque characteristics fulfilled the one horsepower maximum drag constraint. A seal endurance test was also run to examine wear rate. The rotor and seal assembly ran at 205 rpm with a bellows loading of 40 psig for 6 hours. The test indicated a wear of 0.005 inches over the 6 hour period. Assuming a constant, linear wear rate, this corresponds to a seal wear of 0.150 inches over 180 hours; the projected operational life of the WPP.

Figure 20. Pitted Graphite Seal Plate

6.0

WATER CHANNEL TESTS

Tracor expressed a concern that any unburnt diesel fuel that would result from an aborted start attempt would foul the water channel water which would hamper visibility. Some work was done to verify the cold starting capability of the WPP combustor. As a backup it was decided to provide an alternate lightoff system using a fuel which would not foul the channel water. Methanol was chosen. A pressurized bottle, fuel and control system was designed and delivered to the test site.

6.1 SERIES 1

Prototype testing began in August 1989 at Tracor's High Speed Recirculating Water Channel. Solar personnel supported all tests at the Tracor facility. Incorporated into the initial test scheme was a methanol start-up system to facilitate light-off. An ignitor plug, instrumented with a thermocouple, was installed in the rig and acted as a light-off indicator. A P+K (pressure plus a constant) regulator was included in the bellows/seal system to track the combustor pressure and adjust the bellows loading pressure accordingly. Initially, the constant was set at 30 psig, a value based upon rotor/seal component testing at Solar.

Early testing at Tracor revealed that light-off could only be achieved at very low combustor pressures, typically between 4 and 10 psi, with low airflows and heated air. A methanol light-off was easily achieved using the above mentioned parameters, although pump and metering valve problems were encountered when switching to diesel. Modifications were made to the microprocessor, which sends a signal to the fuel metering valve, and a diesel burn was obtained. Diesel runs were short due to the limited fuel supply. A continuous diesel supply was later installed to facilitate future tests. Diesel light-offs were attempted, but were unsuccessful.

In this first series of tests, an airflow discrepancy was revealed. The airflow values indicated by Solar's microprocessor and Tracor's calculations were not in agreement. A constant in the microprocessor was changed to make the two values correspond. However, Solar's data were reduced and the airflows were recalculated using two Solar codes and found to be significantly higher, by 15%-30%, than the values calculated by Tracor. If these higher airflows were correct then the scheduled fuel flow was too low, since the microprocessor signal to the metering valve is dependent upon airflow.

The backpressure imposed by the water column in the rotor channel was also suspect. It appeared to be much less than expected. The original design specified an airflow of 1.1 pps at a combustor pressure of 225 psia with a system pressure drop of 7% which would result in a backpressure of 209 psia minimum. The combustor was designed for this condition, however test results showed that the measured backpressure at the seal did not agree with the airflow being delivered to meet design conditions. Figure 21 shows

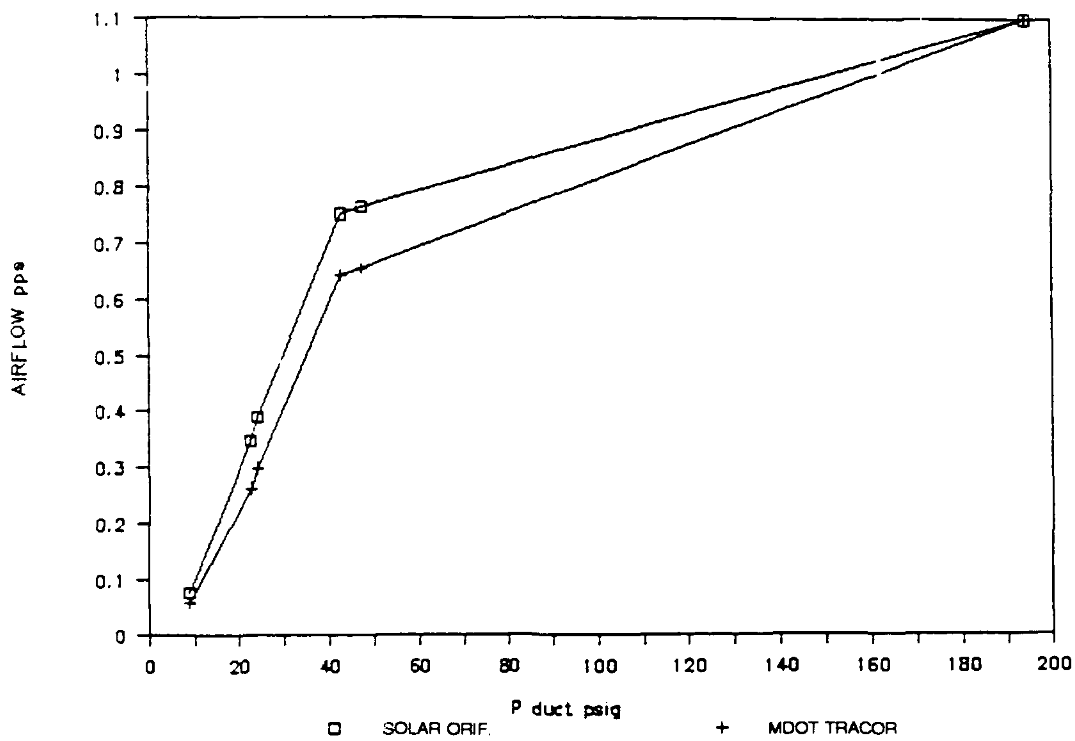


Figure 21. Calculated Airflows vs. Measured Inlet Pressure

the discrepancy where the airflow is plotted against the measured inlet pressure. The 1.1 pps/194 psig was a design point. The curve shows that with the airflows used, the design point could never be reached. It was speculated that the backpressure was substantially different than what was originally proposed. Figure 22 shows the airflow/pressure relationship which Solar estimated based on the design point and compressor characteristics.

System pressure drop was another area of concern. The pressure drop across the combustor sleeve was designed to be 2%. The results during testing at Solar were 2.5%, while at Tracor the measured differential was 9-9.5%. The swirler was designed for a pressure drop of 4%, and during testing at Solar was 4.4%. However, the calculated value at Tracor was approximately 8%. If the calculated airflow was low, as was previously discussed, then this value is also low.

During a shutdown to refill the fuel supply, the chain drive for the rotor was observed to be moving, though the rotor was not rotating. With further inspection, it was seen that the rotor shaft had fractured; the cause of the failure was unknown. Due to the shaft failure, this test series was terminated.

The seal plate appeared to be in good condition with little wear. this is consistent with the data reviewed, as it appeared that a seal did not exist during testing. Almost all measured bellows pressures were below the theoretical bellows settings that were formulated during rotor/seal tests at Solar. An increase in thrust would be expected when

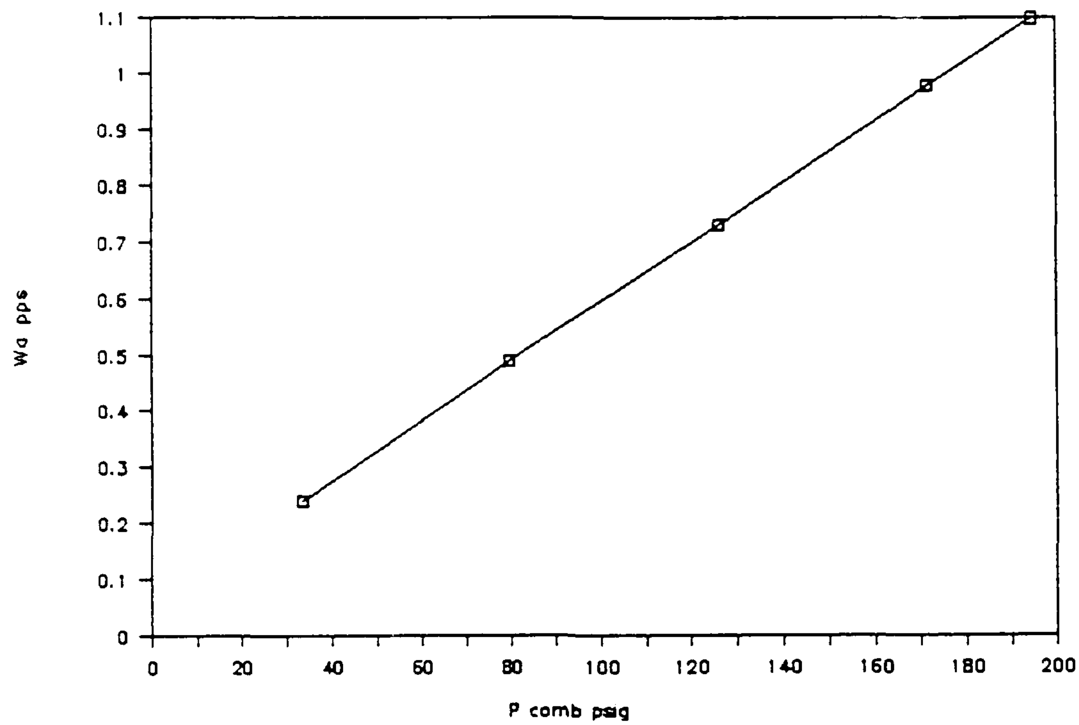


Figure 22. Estimated Combustor Operating Line Given Compressor Characteristics

a seal existed, however no measured rise was indicated by Tracor. No visible evidence of whether or not a seal was occurring could be observed given the structure of the test rig.

Prior to the next series of testing, a flow straightener was sent to Tracor with the intention of being installed at the combustor exit. The effect of swirl on the exit flow of the combustor was a concern. It was discussed that negative pressure may have existed at the center of the trapezoidal rotor inlet due to residual swirl. If that was the case, the gas/water interface would not have provided the effective force needed to expel the water properly. Placing a honeycomb in the combustor exit would have straightened the flow and resolved any questions as to swirl. Due to the swirl present in the combustor, a modified flow may have had a significant effect on rig performance. However, no evaluation tests were performed with the flow :raightener in place.

6.2 SERIES 2

The major change employed in the rig for this second series of tests was the replacement of the simplex fuel nozzle by a dual orifice type. With this nozzle, methanol light-offs were easier to achieve, and the first attempt at a diesel fuel light-off was successful. Although a diesel fuel light was achieved, stable combustion could not be maintained. The rig was disassembled for investigation. Upon teardown, it was observed that the graphite seal plate had sustained severe damage in the direction of rotor rotation, past

the combustor exhaust. Pitting/gouging was concentrated toward the inner and outer edges of the seal plate. Tracor speculated that cavitation may have been the cause of the damage. Having examined the data for the first burning run, the mid-line seal plate thermocouple indicated over 400°F, which was at the top end of the temperature specification on the coated seal plate. Pure Carbon, the seal manufacturer, suggested that if a rapid rise in seal temperature occurred, the coating may have expanded away from the base graphite. This was most likely the case seen during testing.

The rig was reassembled with another simplex fuel nozzle, 7.5 gpm at 70°. Light-offs on diesel were easily achieved using this simplex nozzle and stable combustion was maintained. The damaged seal plate was not replaced for this series of tests. It was believed that both cavitation and high seal temperatures were the cause of the damage to the seal plate. The channel speed and rate of rotor rotation were controlled to avoid the onset of cavitation. Seal temperature did not rise above 190°F for the duration of testing.

The airflows calculated during these tests were more comparable than during previous runs, however a 6% discrepancy remained. Solar's input outlet temperature and Tracor's calculated outlet temperature were in very close agreement.

6.3 SERIES 3

This series of tests was performed to observe operation of the WPP prototype with a transition piece installed at the outlet of the combustor to investigate the effects of combustor outlet port geometry on the WPP performance. The 6 inch long transition piece consisted of a cast stainless steel form, having a circular entrance transitioning into a trapezoidal outlet. The circular inlet coincided with the combustor outlet, while the trapezoidal, transition outlet corresponded with the shape of a single rotor passage. The graphite seal plate was replaced by a bronze plate which was affixed to the outlet of the transition piece. Bronze was used as a cost reduction for test purposes only. A steel plate was attached behind the bronze seal plate to support it through the transients.

Testing with the transition piece installed revealed an improved correlation between Solar's and Tracor's airflows. Higher airflows were achieved, and notably higher combustor pressures were attained, using a different air compressor, compared to previous testing. A high of 115 psig was reached in the combustor. However, light-offs were easily achieved only at low pressures.

Pressure losses across the combustor were also significantly lower than previous testing. Sleeve ΔP was at a high of 3.94% at 115 psig, whereas 10+% was common during Series 1 and 2 testing.

The design air mass flow of 1.0 pps was achieved at the higher pressures, but again, design conditions could not be obtained. Shown in Figure 23, the graph of Solar's airflows better resembles the expected path at design conditions, compared to Series 2

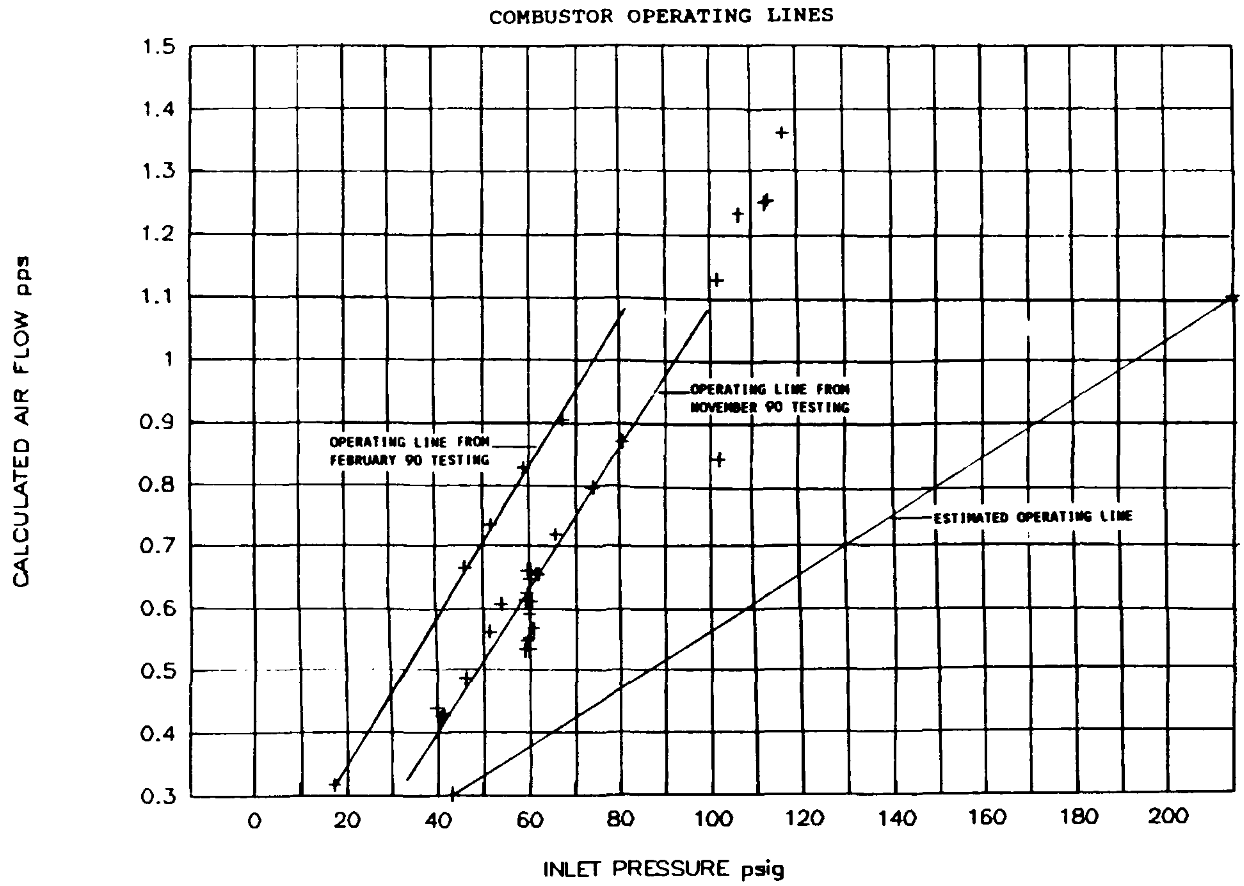


Figure 23. Solar Airflows Compared to Expected Path

testing. However, there still remained a slight airflow discrepancy between Tracor and Solar's calculations.

Included in this series of tests was a B-type thermocouple situated at the exit of the transition zone just ahead of the rotor, installed by Tracor. This thermocouple was specified to measure the outlet gas temperature. Tracor's measured value and Solar's inputted outlet temperature were very close in agreement, +/-150 degrees. Although design conditions specify an outlet temperature of 3000°F, combustion appeared more stable and the water less sooty at a temperature of 2650°F. Tracor supported this view.

Also of major interest was the improvement of the airflows versus combustor pressure. Table 3 shows how the airflows had improved, although design conditions were still unattainable.

During checkout tests, which Tracor ran with the transition piece installed prior to Series 3 testing, a new bellows loading scheme was developed. Tracor had performed cold tests analyzing thrust results compared to bellows pressure and observed that a loading of only 12-15 psig above combustor pressure created a satisfactory seal. Thus, during

Table 3
Combustor Airflows and Pressure Data During In-Water, Prototype Testing

TEST DATE	W (PPS)	Pcombustor (PSIG)
AUG '89	0.64	57.40
AUG '89	0.66	60.60
FEB '90	0.81	56.84
FEB '90	0.82	58.64
NOV '90	0.48	51.20
NOV '90	0.53	59.00
NOV '90	1.015	107.75

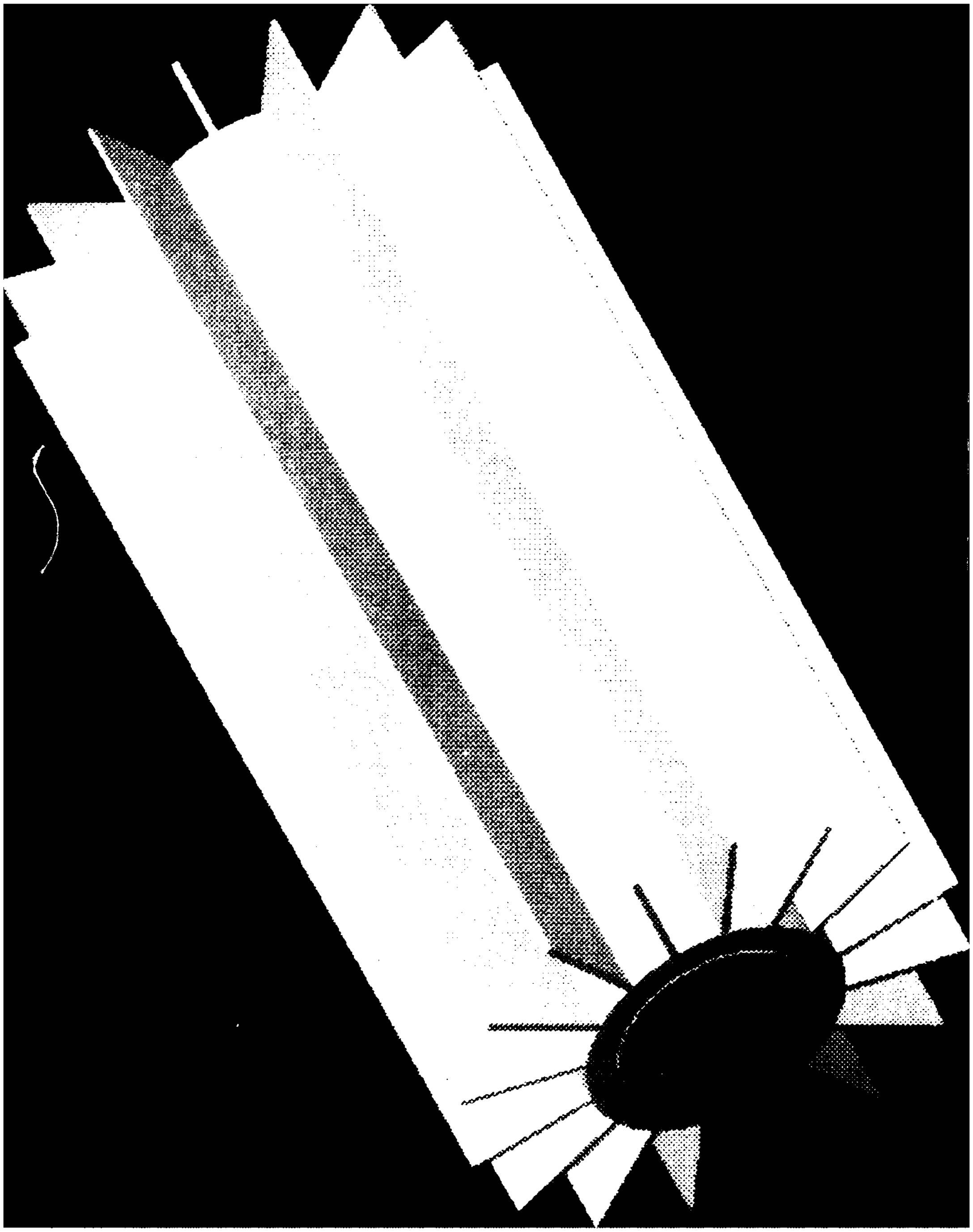
all Series 3 testing, the bellows pressure was maintained at 15 psig above inlet pressure. Bellows loading of 25-35 psig was previously used. This decrease in bellows loading was attributed to Tracor's choice of a bronze seal plate. Solar had investigated the use of bronze for a seal material, but concluded that the poor wear resistance of bronze would not be suitable for the seal plate. In between runs at Tracor, a heavy groove was observed in the seal plate. Using the softer material with poor wear characteristics, the rotor was able to groove itself into the bronze plate establishing a new sealing surface. Roughly 1/32 of an inch was visibly worn away during minimal testing.

As testing continued, lean blow-out runs were performed, at inlet air temperatures settings of 400°and 800°F. These were mainly run to observe thrust data for Tracor. Data were collected and testing concluded.

6.4 MODELLING

In view of the test results at the conclusion of water channel testing, Solar analyzed the seal/rotor interface. It was seen that a single rotor passage spends the majority of the time in an asymmetric mode with respect to the combustor outlet. In other words, the exhaust is unevenly distributed through a single rotor channel. The exhaust and rotor passage are in perfect alignment for only a fraction of the total time that the rotor is passing the seal. For the rest of the time, only part of the channel is exposed to the exhaust expelling only a small percentage of the water.

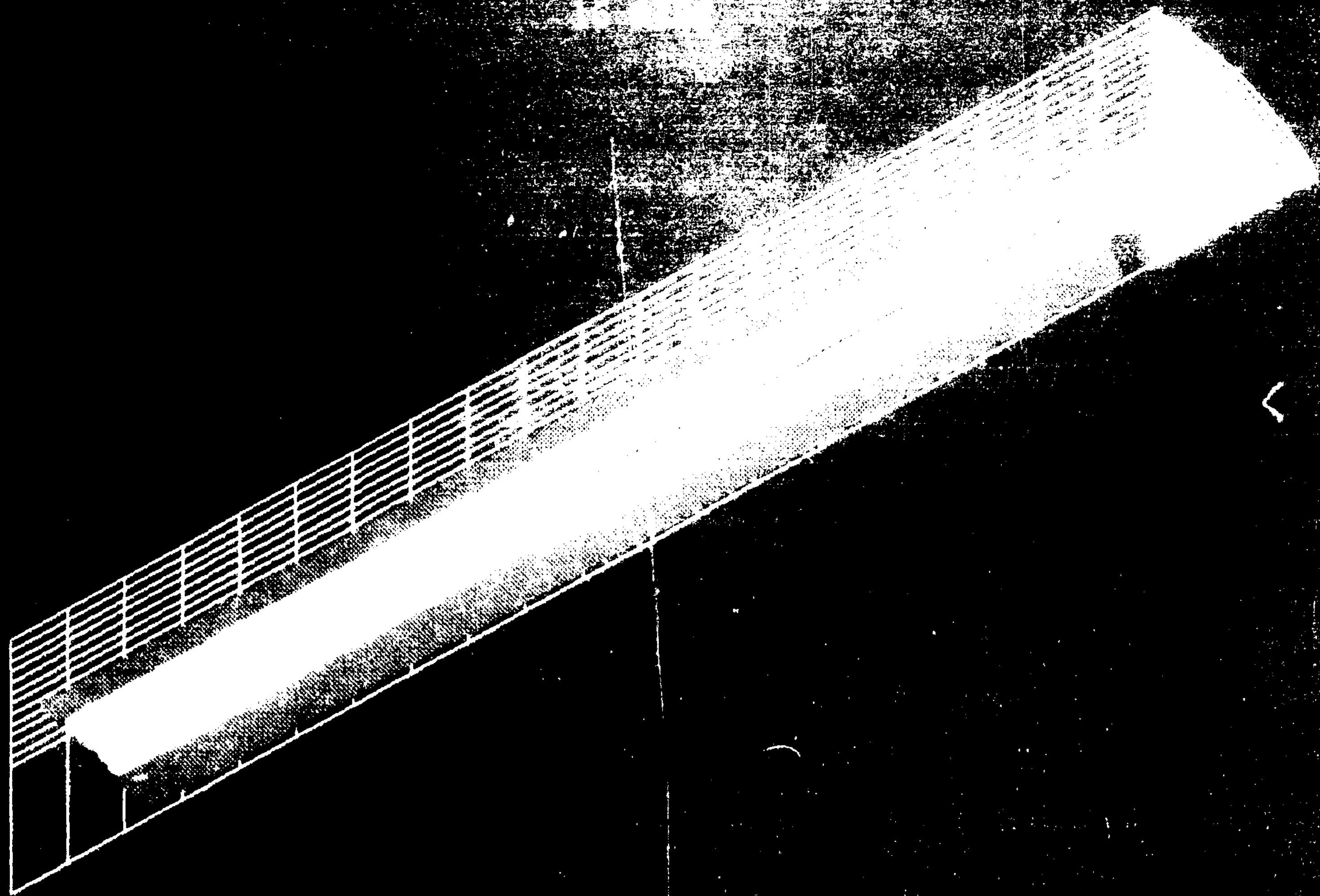
The rotor of the WPP was analyzed at Solar using the Phoenics CFD (Computational Fluid Dynamics) code. The ribs of the rotor, see Figure 24, were modelled as being straight for simplicity. A transient analysis was performed for two rotor positions relative



to the combustor outlet. A modified rotor study was also performed in which case the rotor had 64 ribs. Each model was executed without heat transfer between the combustion exhaust and the water interface.

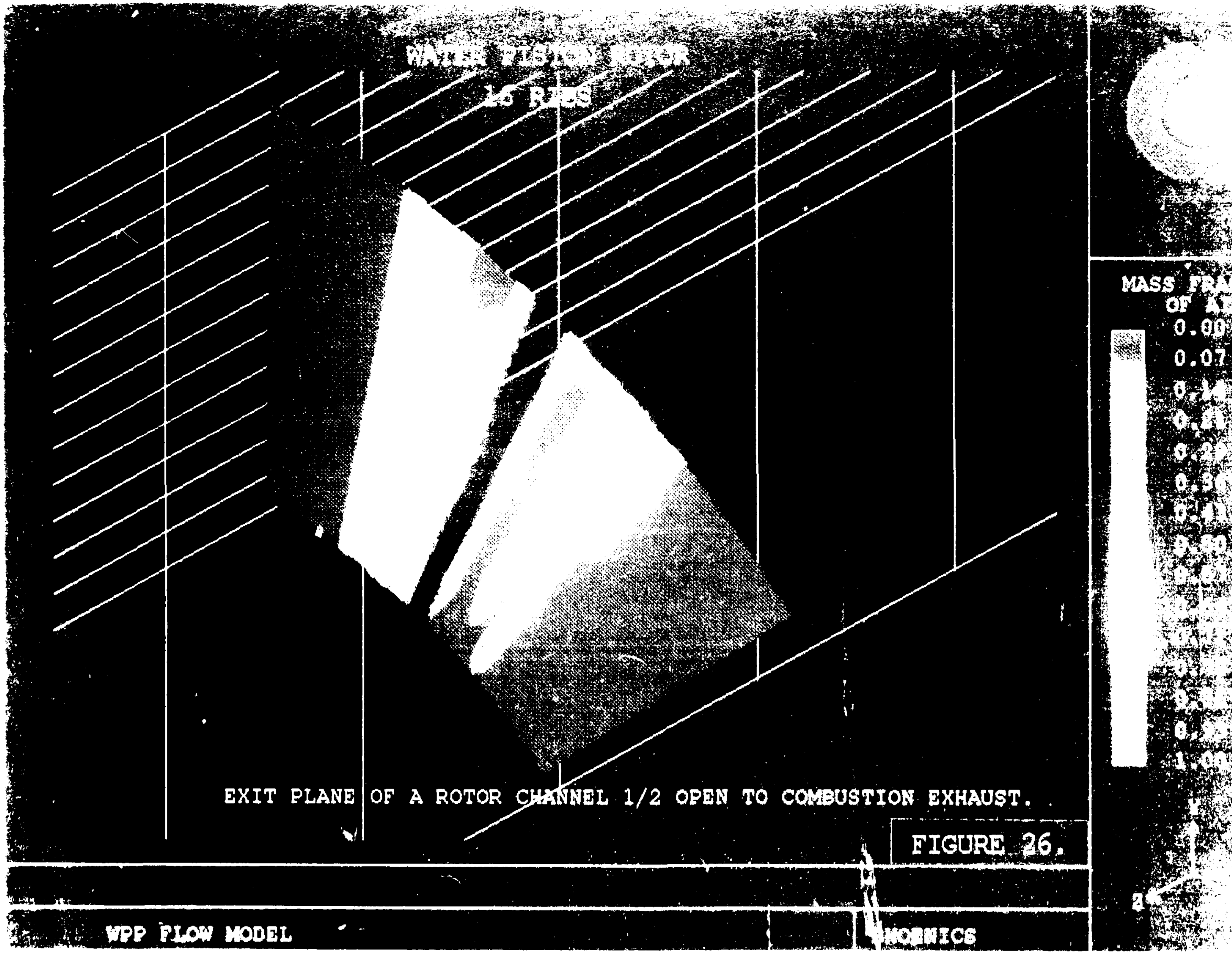
One model was executed for a transient period with the rotor passage 1/2 opened to the combustor exhaust and is shown in Figures 25 and 26. The results here indicate only 1/2 of the forward passage is exhausted, with the hot gas forming a sheet adjacent to the rib and exiting the rotor, while the majority of the channel remains filled with water. Fig. 24 indicates streaks of bubbles over half of the exit plane.

Based on the results of the models it is predicted that the water in the rotor system would only be partially exhausted which would result in low thrust and efficiency. As a possible solution to this problem, a third model was executed with 64 ribs in the rotor. Decreasing the channel opening allows more exposure to the combustor exhaust per opening. The results of this model are presented in Figures 27 and 28. These results indicate nearly 100% of the water is expelled from the rotor during the same period as the model in Figures 24 and 25.



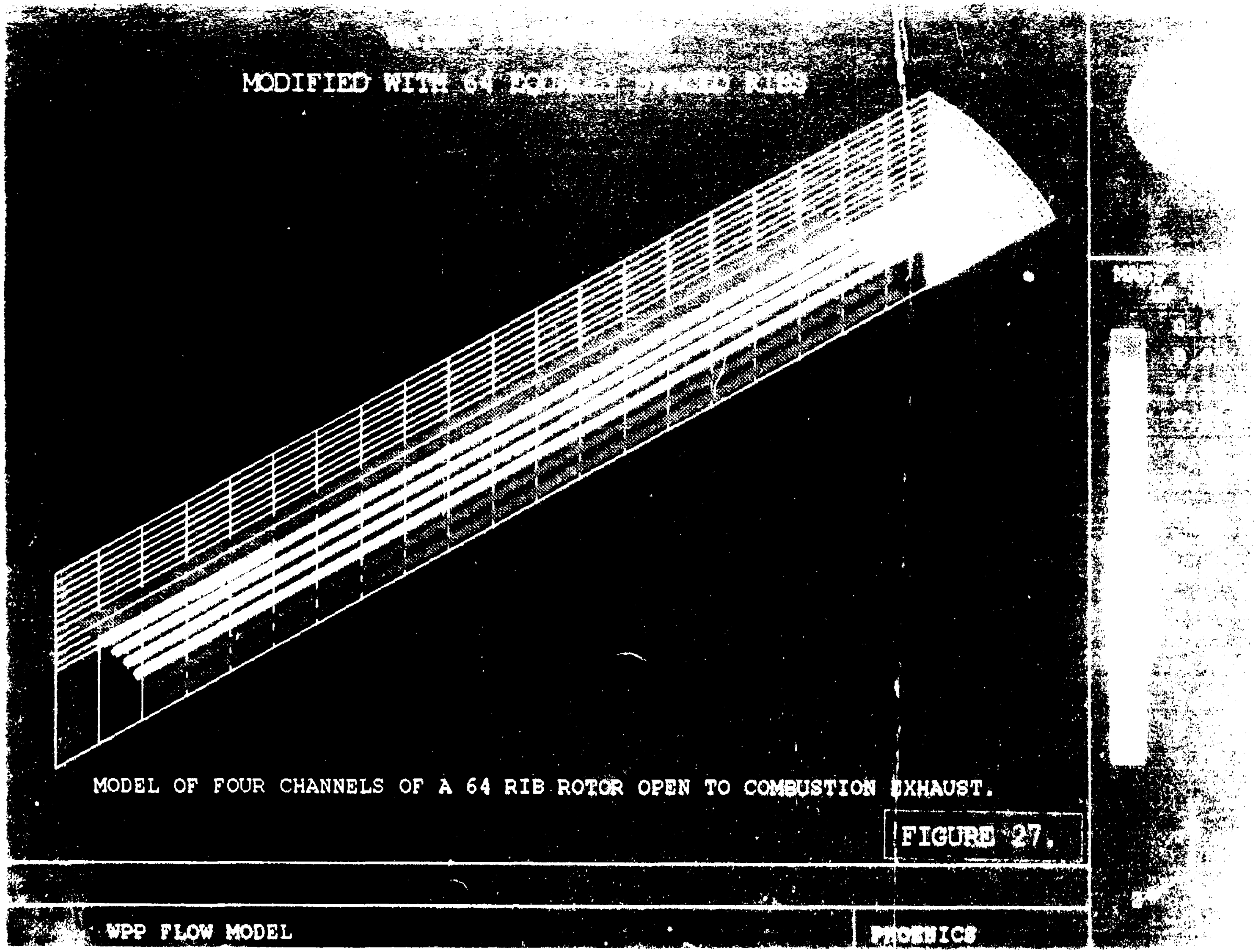
MODEL OF A SINGLE ROTOR CHANNEL 1/2 OPEN TO COMBUSTION EXHAUST.

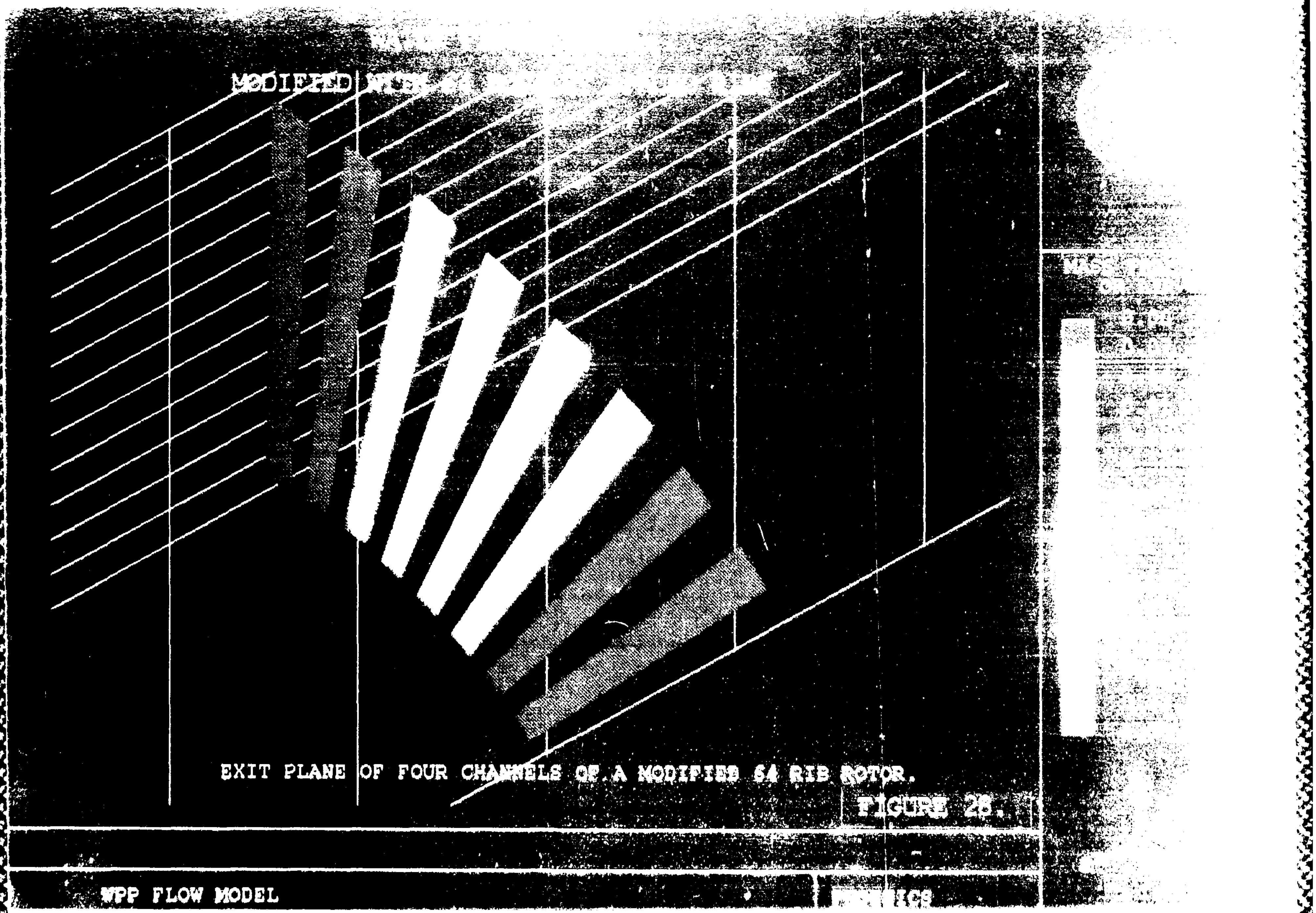
FIGURE 25.



EXIT PLANE OF A ROTOR CHANNEL 1/2 OPEN TO COMBUSTION EXHAUST.

FIGURE 26.


MODIFIED WITH
MODEL OF FOUR CHANNELS OF A 64 RIB ROTOR OPEN TO COMBUSTION EXHAUST.
FIGURE 27.
WPP FLOW MODEL
PHOENICS


MODIFIED
EXIT PLANE OF FOUR CHANNELS OF A MODIFIED 64 RIB ROTOR.
FIGURE 28.
WPP FLOW MODEL

7.0

CONCLUSIONS

The WPP combustor and seal design provided in Task 1 of this project includes all operational features required to meet the original specifications as detailed in Section 3 of this report with only one exception. The combustor diameter requirement of four inches maximum was exceeded. Flanges included for combustor disassembly are the cause of the major diameter increase to 6.0 inches. The combustor liner cooling requirements associated with the 3000°F outlet temperature account for a minor increase in diameter to 4.5 inches.

Water channel testing determined that the original WPP prototype configuration did not exhibit the analytically predicted, propulsive performance. Although the inability to operate the prototype at design conditions was a contributing factor, the rotor design, along with the circular shape of the combustor exhaust, were suspect. Some gains in performance were achieved by going to a modified, trapezoidal combustor outlet port configuration, to conform with the trapezoidal rotor passage. However, overall performance was still below original predictions.

Solar's analytical modelling of the rotor system indicates that the system performance limitations are likely to be associated with the rotor design. Using a CFD code, it was shown that the present configuration of the rotor with 16 channels does not uniformly expel the water. The results indicate that the hot gas exits the rotor channel with the majority of the channel remaining filled with water. Going to a 64 channel rotor would result in nearly 100% expulsion of water from the rotor with each pass through the combustor exhaust.

8.0

RECOMMENDATIONS

8.1 CONDUCT DE-SWIRL TEST WITH FLOW STRAIGHTENER IN THE COMBUSTOR EXHAUST

No evaluation tests were performed with the flow straightener in place. Due to the swirl present in the combustor, modifying the flow may have a significant effect on interface stability/heat transfer and hence overall WPP efficiency.

8.2 RECALIBRATE COMBUSTOR FOR OFF-DESIGN CONDITIONS (I.E. SIGNIFICANTLY LOWER PRESSURES AT DESIGN MASS FLOWS, AND FLUCTUATING BACKPRESSURE

Given the off-design conditions (ie. the significantly lower pressures at design mass flows, and fluctuating backpressure) in which the combustor was operated, it would be desireable to recalibrate the combustor. Efficiency, emissions, and pattern factor could all be reevaluated.

8.3 FABRICATE AND TEST A 64 CHANNEL ROTOR SYSTEM

Decreasing the size of the rotor channel openings results in a more symmetrical exposure to the combustor exhaust. Solar's modelling results indicate nearly 100% of the water is expelled from the rotor during each channel pass using a 64 channel rotor. Water channel testing would verify this approach.

9.0

REFERENCES

1. Lefebvre, A. H. and Herbert, M. H., "Heat-Transfer Processes in Gas Turbine Combustion Chambers", Vol. 174, No. 12, 1960 Proc. Instn. Mech. Engrs.

2. Marek, Cecil J. and Tacina, Robert R., "Effects on Free-Stream Turbulence on Film Cooling", NASA TN D-7958, June 1975.

3. Kercher, D. M., and Tabakoff, W., "Heat Transfer by a Square Array of Round Air Jets Impinging Perpendicular to a Flat Surface Including the E..ects of Spent Air", ASME Paper No. 69-GT-4, December 17, 1969.

4. Paxton, R. R., et al., "Wear Rates of Impregnated Carbons Sealing Warm Water". J. Am. Soc. Lubrication Engineers.

5. Tuthill, A. H. and Schellmoller, C. M., "Guidelines for Selection of Marine Materials", Ocean Science and Ocean Engineering Conference, Marine Technological Society, Washington, DC, June 14-17, 1965.

6. Sedicks, A. J., Corrosion of Stainless Steels, John Wiley, New York, 1979, p. 154.

7. "Protective Finishes for Space Vehicle Structures and Associated Flight Equipment", MSFC-SPEC-250A.

8. "Handbook of Huntington Alloys", Brochure, January 1968.

9. Sedicks, A. J., op. cit. p. 156.

APPENDIX A

Impingement Cooling Flow & Heat Transfer Analysis

Table A-1

IMPINGEMENT COOLING FLOW & HEAT TRANSFER ANALYSIS
KERCHER-TABAKOFF CORRELATION

FLOW SOLUTION

P1	P2	P2/P1	TEMP	UP DENS
222.75	217.125	.974747	800	.477675

Y1-ORIF
.992415

NO HOLES	DIA-IN	A2-SQ IN	L/D	W-LB/SEC
42	.085	.238208	.470588	.170196
42	.08	.211008	.5	.152331
42	.075	.185456	.533333	.135447
42	.07	.161553	.571429	.119546
42	.065	.139298	.615385	.104626
42	.06	.118692	.666667	.906875E-01
42	.057	.10712	.701754	.827957E-01
42	.054	.961405E-01	.740741	.752574E-01
42	.049	.079161	.816327	.634788E-01
42	.045	.667643E-01	.888889	.547627E-01
42	.045	.667643E-01	.888889	.547627E-01
-----	-----	-----	W TOTAL	1.10389

G-LB/HR-SQ FT	NRE	CD
370388	32389.9	.656765
374243	30801.9	.6636
378612	29213.9	.671347
383605	27625.9	.6802
389360	26037.8	.690415
396087	24449.8	.702333
400686	23497	.710488
405796	22544.2	.719548
415702	20956.2	.737114
425213	19685.8	.753978
425213	19685.8	.753978

NOTES:FLOW
(1) CD SOL'N FOR .25<L/D<2 NOT VALID FOR NRE<1E4
(2) FLOW BASED ON ULT CD WHEN .25<L/D<2
(3) NO SOL'N PRODUCED FOR L/D>10
(4) FOR 2<L/D<10 CD=F(NRE-M)
(5) NRE-M=NRE/(C/M)

HEAT TRANSFER SOLUTION

DIA.-FT	XN/D	ZN/D	THETA	GX/GJ
.708333E-02	2.94118	2.94118	90	.301752
.666667E-02	3.125	3.125	90	.21281
.00625	3.33333	3.33333	90	.134915
.583333E-02	3.57143	3.57143	90	.674381E-01
.541667E-02	3.84615	3.84615	90	.977281E-02
.005	4.16667	4.16667	90	0
.00475	4.38596	4.38596	90	.382294E-01
.0045	4.62963	4.62963	90	.807746E-01
.408333E-02	5.10204	5.10204	90	.117023
.00375	5.55556	5.55556	90	.145892
.00375	5.55556	5.55556	90	.172985

ANG. FACT.	PHI-1	PHI-2	M	(NPR)^1/3
.996727	.171556	.709036	.694739	.881198
.996727	.14486	.76438	.704793	.881198
.996727	.120989	.826843	.715656	.881198
.996727	.998045E-01	.898028	.727454	.881198
.996727	.811624E-01	.980149	.740344	.881198
.996727	.649185E-01	.997009	.754523	.881198
.996727	.562623E-01	.925959	.763751	.881198
.996727	.483844E-01	.85132	.7736	.881198
.996727	.368955E-01	.783055	.791621	.881198
.996727	.290931E-01	.727259	.807761	.881198
.996727	.290931E-01	.692468	.807761	.881198

(NRE)^M	(ZN/D)^.091	NNU	H-B/HR-SQ FT-F
1360.06	1.10315	160.294	685.681
1457.2	1.10925	157.202	714.482
1569.74	1.11579	153.897	746.092
1701.59	1.12282	150.402	781.232
1858.09	1.13041	146.758	820.981
2046.69	1.13868	132.486	802.868
2179.54	1.14401	114.091	727.781
2330.63	1.14965	96.9444	652.759
2635.38	1.15986	77.5647	575.562
2942.03	1.16888	63.9066	516.366
2942.03	1.16888	60.8494	491.663

NOTES:HEAT TRANSFER
(1) NO SOL'N. PRODUCED FOR NRE<300 OR >1E5
(2) TRUE UPPER LIMIT OF SOL'N. IS NRE=3E4
THIS UPPER LIMIT HAS BEEN EXTRAP. TO 1E5
(3) OTHER LIMITS: (A) 3.13<XN/D<12.5, (B) 1<ZN/D<4.8
(C) 3E2<NRE<3E4, (D) .263<L/D<1.18,
(E) 15<THETA<90

SAMPLE OUTPUT
NOMENCLATURE

P1= PRESSURE UPSTREAM OF IMPINGEMENT HOLES

P2= PRESSURE DOWNTREAM OF IMPINGEMENT HOLES

TEMP= INLET AIR TEMPERATURE F

UP DENS= DENSITY OF AIR UPSTREAM OF IMPINGEMENT HOLES

Y1-ORIF= ORIFICE EXPANSION FACTOR

NO HOLES= NUMBER OF IMPINGEMENT HOLES EACH ROW

DIA-IN= DIAMETER OF IMPINGEMENT HOLES EACH ROW

A2-SQ IN= AREA OF IMPINGEMENT HOLES EACH ROW

W-LB/SEC= FLOW RATE OF IMPINGEMENT HOLES EACH ROW

G-LB/HR-SQ FT= MASS FLUX OF IMPINGEMENT HOLES EACH ROW

NRE= REYNOLDS NUMBER OF IMPINGEMENT HOLES EACH ROW

CD= ORIFICE DISCHARGE COEFFICIENT

DIA-FT = DIAMETER OF IMPINGEMENT HOLES EACH ROW

XN= CENTER-TO-CENTER HOLE SPACING

ZN= DISTANCE BETWEEN IMPINGEMENT PLATE AND LINER

THETA= ANGLE BETWEEN IMPINGMENT JET AND LINER

GX= AIR FLOW RATE PER UNIT CROSSFLOW AREA UPSTREAM OF JETS

GJ= AIR FLOW OF JETS PER UNIT AREA OF IMPINGMENT JETS

ANG. FACT.= EFFECT OF JET ANGLE ON IMPINGEMENT COEFFICIENT

PHI-1= CORRELATING CONSTANT (REFERENCE 3.)

PHI-2= CORRELATING CONSTANT (REFERENCE 3.)

M= CORRELATING CONSTANT (REFERENCE 3.)

NPR= PRANDTL NUMBER

REN= REYNOLDS NUMBER

NNU= NUSSELT NUMBER

H-B/HR-SQ FT-F= AVG. IMPINGEMENT HEAT TRANSFER COEFFICIENT

APPENDIX B

Production Cost Analysis

SOLAR TURBINES INCORPORATED

INTEROFFICE MEMO

M. E. Flinn

TO: [redacted]
P. B. Roberts

FROM: M. E. Flinn

SUBJECT: WATER PISTON PROPULSION COST ANALYSIS

DATE: February 22, 1988

Cost Accounting has performed a cost study for the Water Piston Propulsion System. This study was performed with the following assumptions:

- Requirements will be 133 pieces per year starting in 1990 and ending in 1992.

- Part numbers 171134-100-2, -3, and -4 will be produced from a single piece of tubing rather than using a "form and weld" process. The use of a single piece of tubing will reduce distortion, time, and cost. In the event the Navy will not accept this "single piece tubing" concept, the cost will be higher by an unknown amount (manufacturing engineering did not develop production routings for the "form and weld" concept).

- $500,000 will need to be invested in a small Okuma machining center. This investment would be amortized over the life of the asset as opposed to billing the Navy for it directly.

The following chart shows our projected prices to the Navy for recurring and non-recurring items in 1988 and 1990 dollars. These prices include ERMA at 24.5% of product cost, 15% profit on the product, and 20% profit on tooling. The combustor prices also include contingencies of approximately 8%.

DESCRIPTION	1988 PRICE	1990 PRICE
Price Per Combustor	$ 24,590	$ 26,780
Price of Non-Recurring Items	400,470	436,090

Please contact me, on extension 5721, for a complete cash-flow study and return-on-investment analysis prior to entering into any pricing negotiations or agreements with the Navy.

cc: Dick Cruz
Frank Fabro

7329g/c

WATER PISTON PROPULSION SYSTEM
NON-RECURRING PRICE SUMMARY

NON-REC2

DESCRIPTION	AMOUNT
TOOLING (NOT INCLUDING FACILITIES)	$310,871
MFG ENGINEERING	62,622
NC PROGRAMMING	17,970
TOTAL NON-RECURRING (NOT INCLUDING FACILITIES)	$391,463
INFLATION TO 1988 @ 2.3%	9,004
PROJECTED COST (1988 $)	$400,467
INFLATION TO 1990 @ 11.4%	44,627
PROJECTED COST (1990 $)	$436,090